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1.5 MILLION POUND LOAD CELL CALIBRATION AND H-AREA THRUST MEASURING SYSTEM

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Prepared for

National Aeronautics and Space Administration

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TECHNOLOGY REPORT

1.5-MILLION POUND LOAD CELL CALIBRATION
AND
H-AREA THRUST MEASURING SYSTEM

Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

27 June 1966

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ABSTRACT

This report describes the 1.5 million pound load cell calibration system and the Test Stand H-8 1.5 million pound thrust measurement system developed for the M-1 engine test complex.

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SUMMARY

This report is presented in two parts; Part One deals with the force cell calibration system and Part Two deals with the thrust chamber test stand. Calibration of force measuring cells to 1.5 million lb presents problems in both techniques and funding. A number of methods were considered before a Force Multiplying System was selected. Each of these methods is discussed in this report. Addition of a Force Multiplying System to the existing 60,000 lb National Bureau of Standards calibrated dead-weight machine at Aerojet-General Corporation appeared to be the best over-all approach. The completed system has an accuracy of better than $\pm$ 0.10% and is capable of applying loads in either tension or compression.

The thrust measurement system on Test Stand H-8 is designed to support the M-1 thrust chamber assembly in a horizontal attitude and to measure axial thrust during a test firing. This thrust measurement system utilizes a single 1.5 million lb load cell and is equipped with an integrally-mounted calibration system. Using appropriate data reduction techniques, the system is capable of a static force measurement accuracy of $\pm$ .5% 3σ and a transient accuracy of $\pm$ 2% 3σ.

PART I

1.5-MILLION POUND FORCE CALIBRATION SYSTEM

I. INTRODUCTION

With the development of high thrust rocket propulsion systems, it is necessary to have the means for calibrating thrust measuring devices with full-scale ranges to 1.5-million lb and higher. The most reliable type of calibration would be the application of forces by dead weight; however, the use of dead weight in the thrust range of 1.5 million lb is extremely costly.

Input calibration forces can be applied by other means. Some of these are:

A. Calculation of forces from a known area of a hydraulic piston and a known applied pressure.

B. A combination of lower range cells set in parallel.

C. A transfer standard calibrated by the National Bureau of Standards.

D. Hydraulic captive pressure bellows.

E. A multiplication arm loaded by dead weight (Force Multiplying System).

Each of these methods will be subsequently discussed in this report.

A Force Multiplying System was determined to be the best method for load application and a detailed specification of the requirements was prepared. The completed system was to have a system accuracy of $\pm$ .15%. The complete specification is presented as Appendix A.

II. PROPOSED METHODS

A. FORCE APPLICATION BY HYDRAULIC PISTON

The simplest method of force application is to supply hydraulic pressure to a piston of a known area. However, this method will not provide the required accuracy for the calibration of thrust cells as it depends directly upon the ability to measure the applied pressure and the effective area of the piston. In addition, the frictional losses in the piston are undefined.

B. COMBINATION OF LOWER RANGE CELLS

This method of loading poses many problem areas. The most significant would be in obtaining a series of cells which exhibit the same deflections for a given load. It has been found that if equal deflections are not observed on each load cell the load distribution will shift. With this phenomenon present, the accuracy of the applied load would be uncertain.

A second problem arises in the difficulty of mounting such a system for calibrating a cell in tension.

C. TRANSFER STANDARD FROM THE NATIONAL BUREAU OF STANDARDS

Two types of Transfer Standards from the National Bureau of Standards are available. These are a load cell calibrated with an associated readout or a Morehouse proving ring.

At the time the methods for loading were being discussed, the capability of the National Bureau of Standards was limited to 110,000 lb by dead weight. Higher range devices were available, but with a decline in accuracy.

D. HYDRAULIC CAPTIVE PRESSURE BELLOWS

The use of captive pressure bellows to determine force loads have been successful in the past. This device consists of a cylindrical bellows closed at both ends with a provision for applying internal pressure from a dead-weight tester. However, this system is sensitive to temperature and care must be taken to maintain the device at a constant temperature.

With this type of loading, the time required to ensure an accurate measurement would not be desirable for a continuous method of thrust cell calibration.

E. FORCE MULTIPLICATION SYSTEM

Demonstrations have shown in the past that the use of dead-weight load applied to a lever arm system can produce accuracy and ease of operation. With the available weight of 60,000 lb, this method of load application seemed the most desirable.

Two types of lever arm pivots could be used for this type of system; the knife edge and the flexure pivots.

The knife edge pivots become prohibitive in this application because of the length required. For a knife edge pivot to carry a load of 1.5 million lb at a working force of 6000 lb/in., the required length would be approximately 20 ft. Two associated factors would limit the confidence in such a system. These are the ability to place the knife edges and pivots parallel over the 20 ft length and the unknown frictional forces which could cause non-repeatability. In addition, the temperature differential across the length of the knife edge must be held to approximately 3°F to maintain alignment.

Flexure pivots appeared to fit all of the requirements of the desired system.

Data associated with several manufacturers of flexures have demonstrated capabilities of proven designs offering not more than 0.15% transverse reactions to bending extremes approaching one degree of angular deflection. It appeared very feasible that these errors could approach less than 0.01% if the beam was balanced to less than 30 sec of arc. This factor, in addition to the intrinsic characteristic of flexures having no friction, favored design concepts of this nature.

1. Double Arm Beam Calibrator

The double arm beam concept is shown in Figure No. 1. This concept was capable of loading in tension and compression. However, with the loading yoke design, it was necessary to counterbalance the beam for tension loading (the cell is mounted the same for tension and compression). With the load of 60,000 lb applied to the loading beam, the smallest load that could be applied in tension would be 250,000 lb by removing the first weight increment of 10,000 lb. This limited the use of the calibrator. The proposed system incorporated a leveling device to relevel the beam after each loading change to eliminate any cosine errors. The weights were to be loaded using a hydraulic pickup system.

2. Single Arm Beam Calibrator

A second proposal for a force multiplication system was a single arm beam (Figure No. 2). Unlike the double arm beam, the unit being calibrated is not mounted in the same position for tension and compression loading. This allows the force multiplying system to apply a load of 25,000 lb as the smallest increment for tension as well as compression loading. In addition, this system has a self-contained hydraulic system mounted on it to eliminate any errors resulting from the

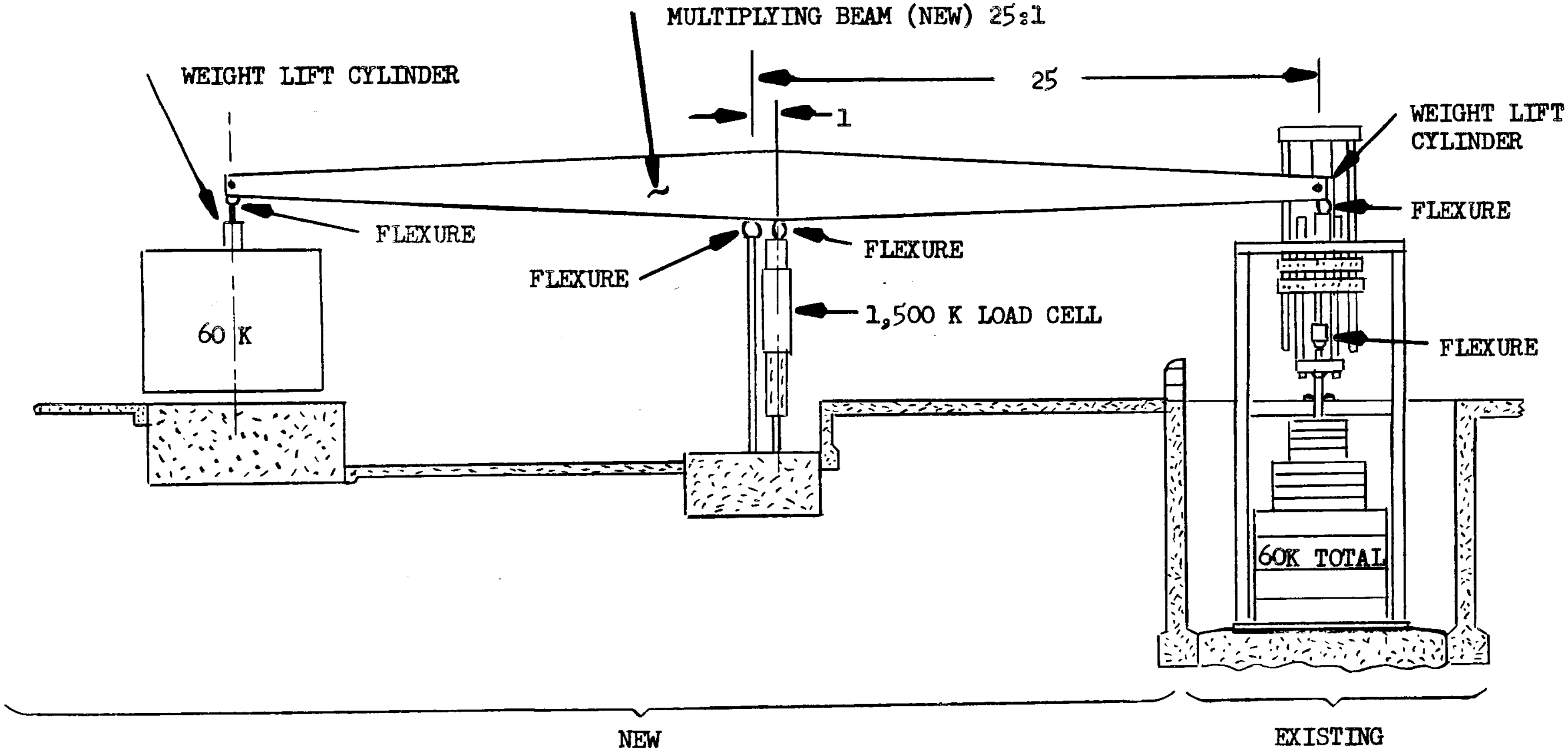


Figure 1

DOUBLE BEAM CALIBRATOR

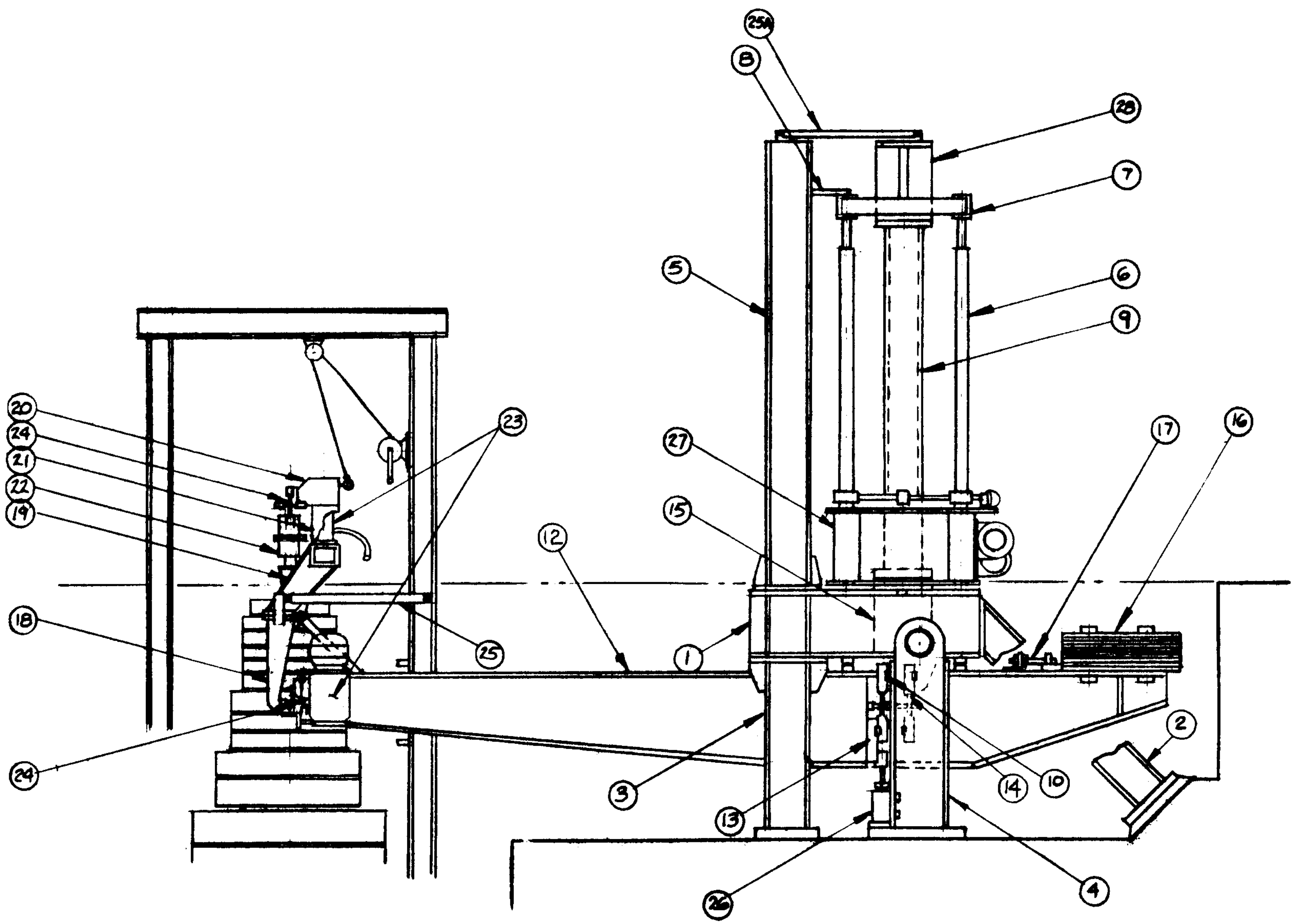


Figure 2

SINGLE BEAM CALIBRATOR, 1.5-MILLION POUND FORCE CALIBRATION SYSTEM changing volume of the weight pickup piston. It also incorporates an automatic leveling system. This type of force multiplying system was selected as the best approach to meet the requirements.

III. TECHNICAL DISCUSSION

A. DESCRIPTION OF SYSTEM COMPONENTS

The force multiplying system was used for the calibration of thrust measuring cells with a full scale of up to 1.5 million lb. This system incorporates a dead-weight pick-up system multiplying beam, a load yoke, a traveling beam, a grounding frame, and a destabilization system. Associated equipment and systems include a weight pick-up hydraulic system, a control system, 60,000 lb of dead-weight and a leveling system.

In the following descriptions of the components, the numbers in parentheses are callouts in Figure No. 2 unless otherwise specified.

1. Grounding Frame

The grounding frame consists of a frame (1), two braces (2), two front supports (3), two main columns (4), and two auxiliary support beams (5). The entire frame is fixed in position by J-bolts that are set into the foundation. Four hubs are provided in the grounding frame for the attachment of the ball screw jacks (6) which transmit the force from the beam through the test cell to the frame. At the bottom of each hub is an end plate with a screw in the center to allow for the take-up of clearance in the axial anchoring of the individual screw jacks. The ball screw jacks are anchored torsionally at the bottom in the ground frame hubs and on top in the top jack support frame (7). This support frame is steadied by support ties (8) to the auxiliary support beams (5).

The connection between the frame (1) and the main support columns (4) is made by two pins (9). Machined surfaces on the inside face of the main columns (4) provide the mounting surfaces (1) for the main tension (pivot) flexures (11). Figure No. 3 shows the ground frame set over the force multiplying beams without the braces (2) and the ball screw jacks (6) installed.

2. Force Multiplying Beam

Applied dead weight is multiplied by a 25:1 ratio beam (12). The beam is attached to the ground frame by two cross-tied plate tension flexures (11). These main support tension flexures are keyed and bolted to both the main column (4) of the ground frame and the multiplying beam outer extensions (13). Figures No. 4 and No. 5 are two views of the force multiplying beam. Figure No. 4 shows the outer extension of the beam with the flexure keys in position.

The multiplied output force is transmitted from the multiplying beam through the two cross-tied compression plate output flexures (14) to the

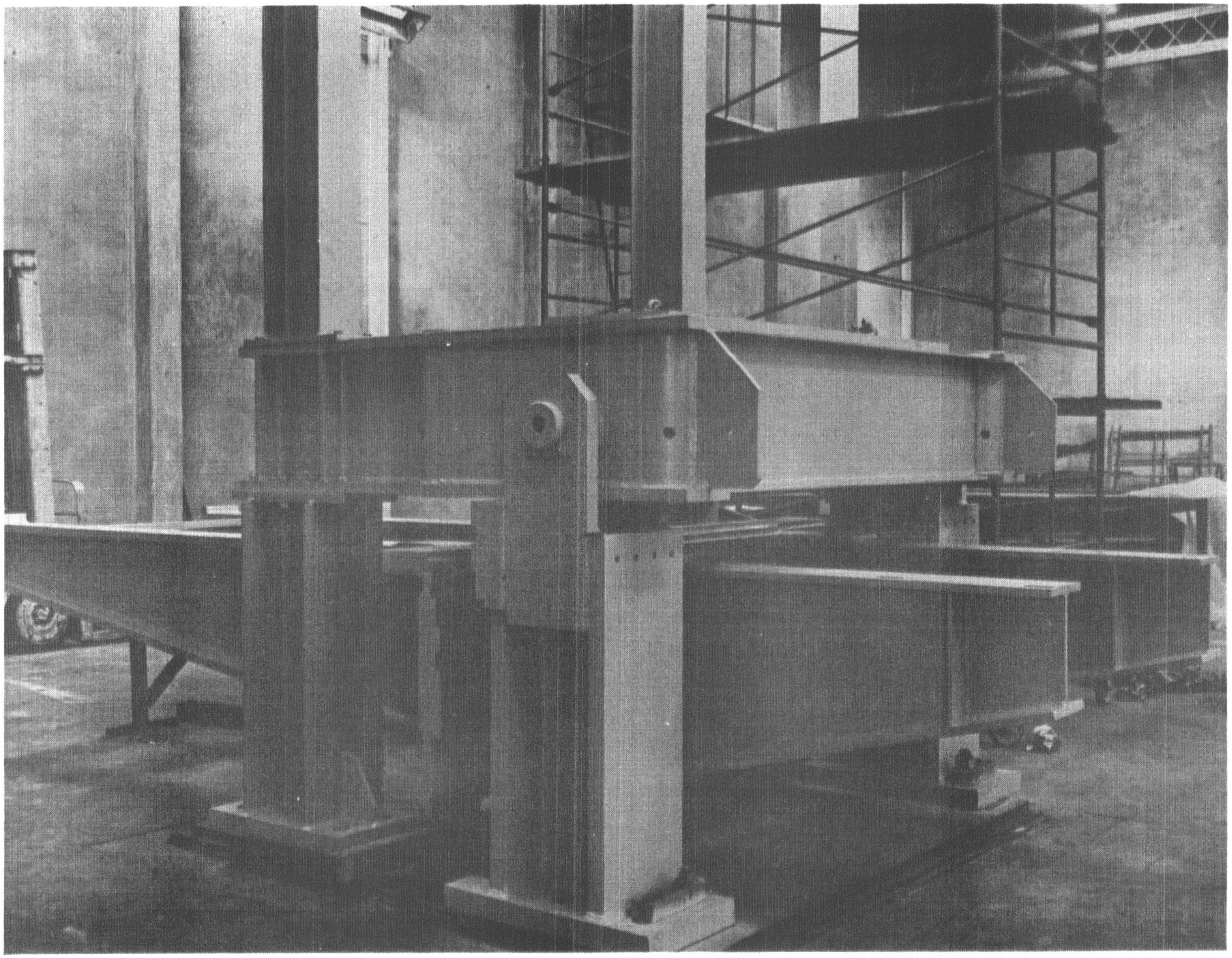


Figure 3

GROUND FRAME AND MULTIPLYING BEAM

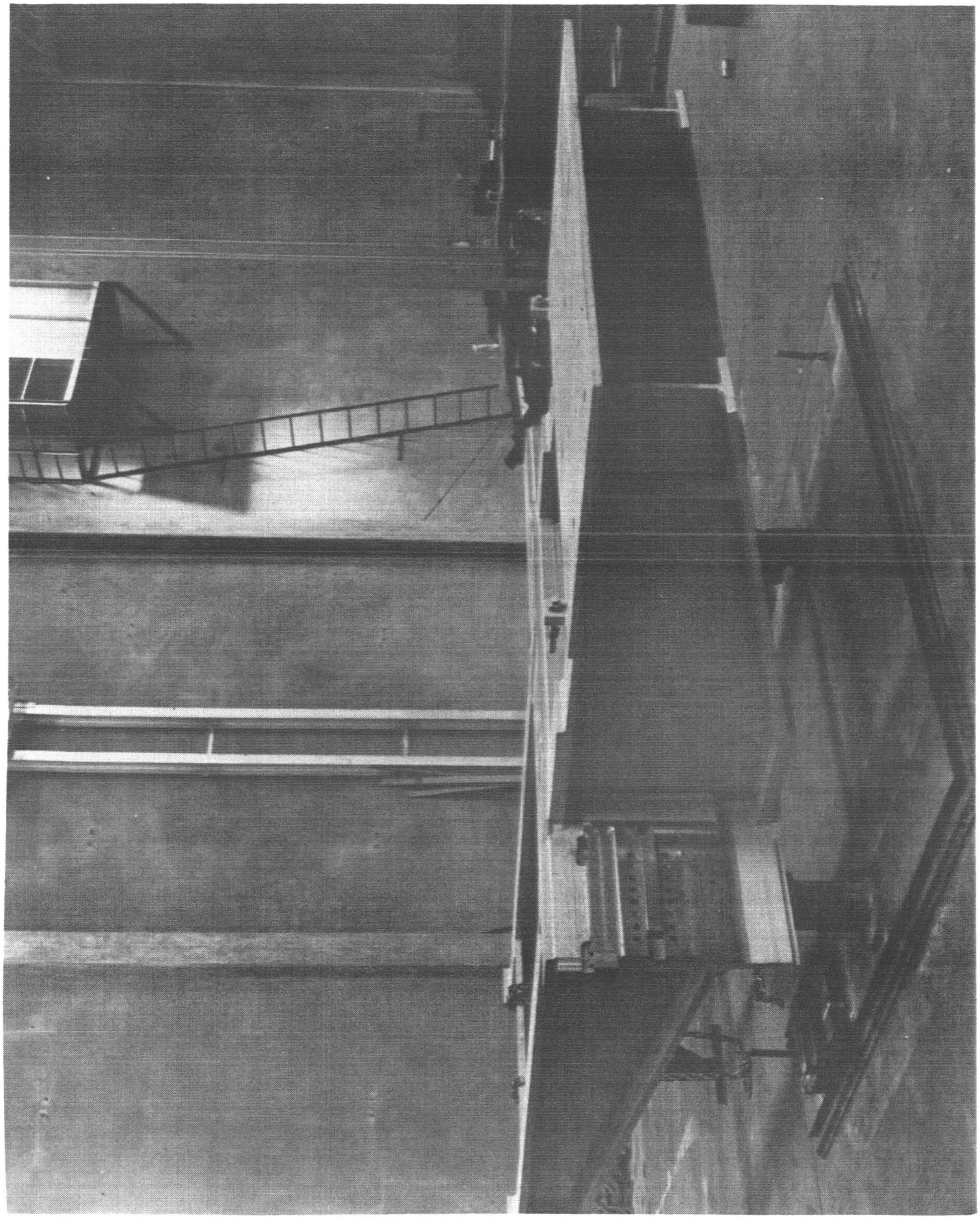


Figure 4
MULTIPLYING BEAM (THREE- QUARTER VIEW)

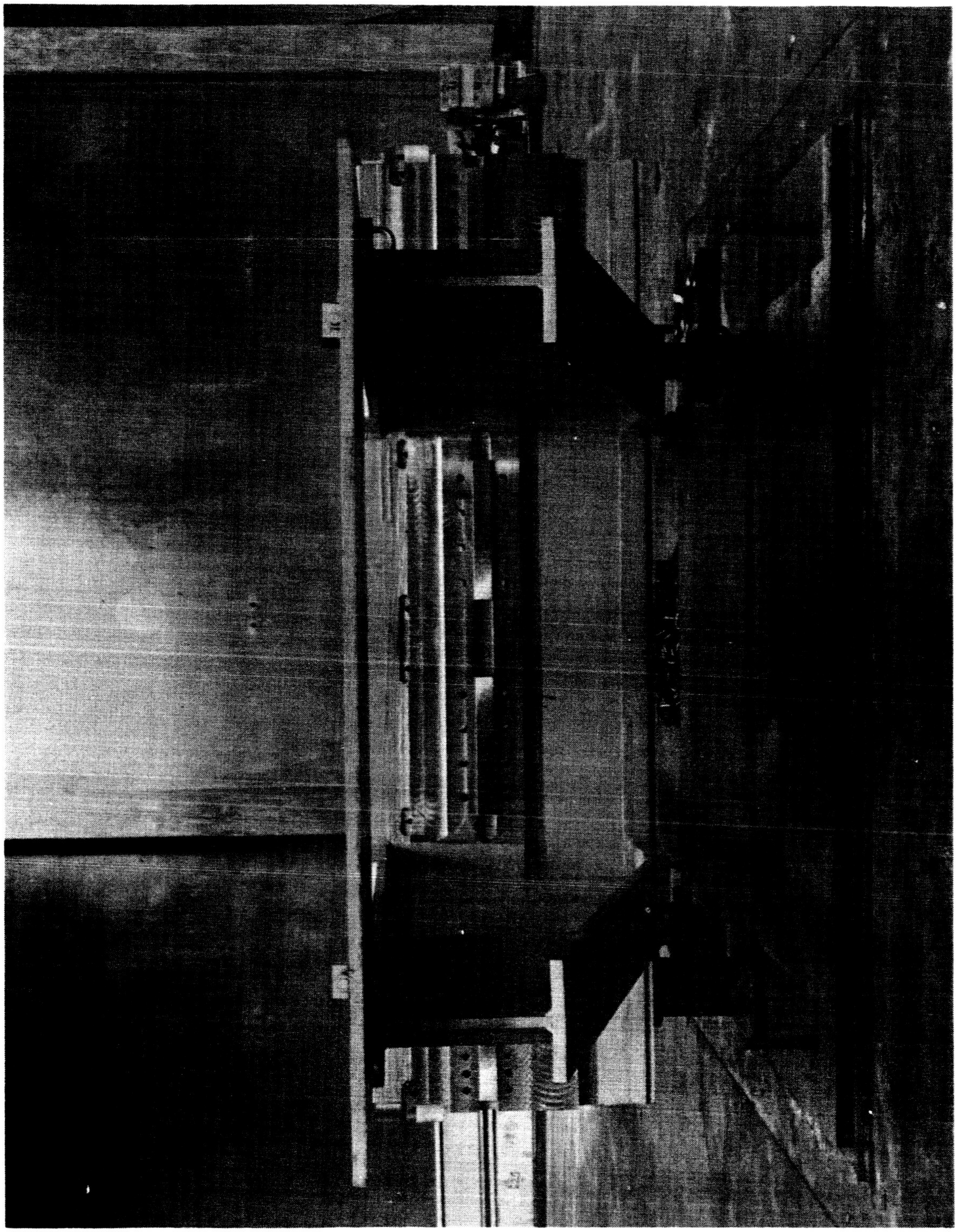


Figure 5
MULTIPLYING BEAM (END VIEW)

lower load yoke beam (15). These flexures are keyed and bolted on the inboard side of the beam. The surfaces with the keys in place are shown on Figure No. 5.

The lines of action of the main support (pivot) flexures and the plate output flexures is six inches. This short arm length in conjunction with a long arm length of 150-in. provides a multiplication ratio of 25:1.

A counterbalance weight, consisting of a stack of steel plates (16), is bolted through slotted holes to the rear extension of the beam. This weight is used to balance the beam in a no-load condition for a level position. The adjusting screws (17) are attached to the counterbalance weights to give a fine adjustment of the beam balance.

3. Pivot and Loading Flexures

This system contains four main flexures; two are main support (pivot) and two are compression loading flexures. In addition, each flexure has two cross-tie flexures. Figure No. 6 shows a flexure and its cross-ties. The flexures are mounted so that the center line of the flexures and cross-ties are in a horizontal line when the beam is in a level condition.

The material selected for the flexures was Vasomax 300, which has a tensile strength of 270,000 psi and is designed to work at 190,000 psi.

A scale mockup of the system was fabricated and tested for a period of 25,000 cycles deflection at full load with no failures. This exceeded the specification of 10,000 cycles.

It was anticipated that it may be necessary to replace the flexures after a one-year period but to date no wear has been detected.

4. Weight Pick-Up System

The weight pick-up system consists of two pick-up columns (18), a weight pick-up support (19), a pick-up beam (20), a flexure holder assembly (21), a weight pick-up stud (22), and a hydraulic system (23).

The dead-weight load is picked up by a single-acting hydraulic ram or piston which is part of the hydraulic system (23).

To prevent interference with the 60,000 lb dead-weight machine, the hydraulic ram is mounted eccentrically from the center line of the weight stack on the pick-up support (19).

The connection between the dead weight and the multiplying system is made through cross-tied tension flexures (24). There are four weight pick-up tension flexures in the weight pick-up system. Two of the flexures carry the

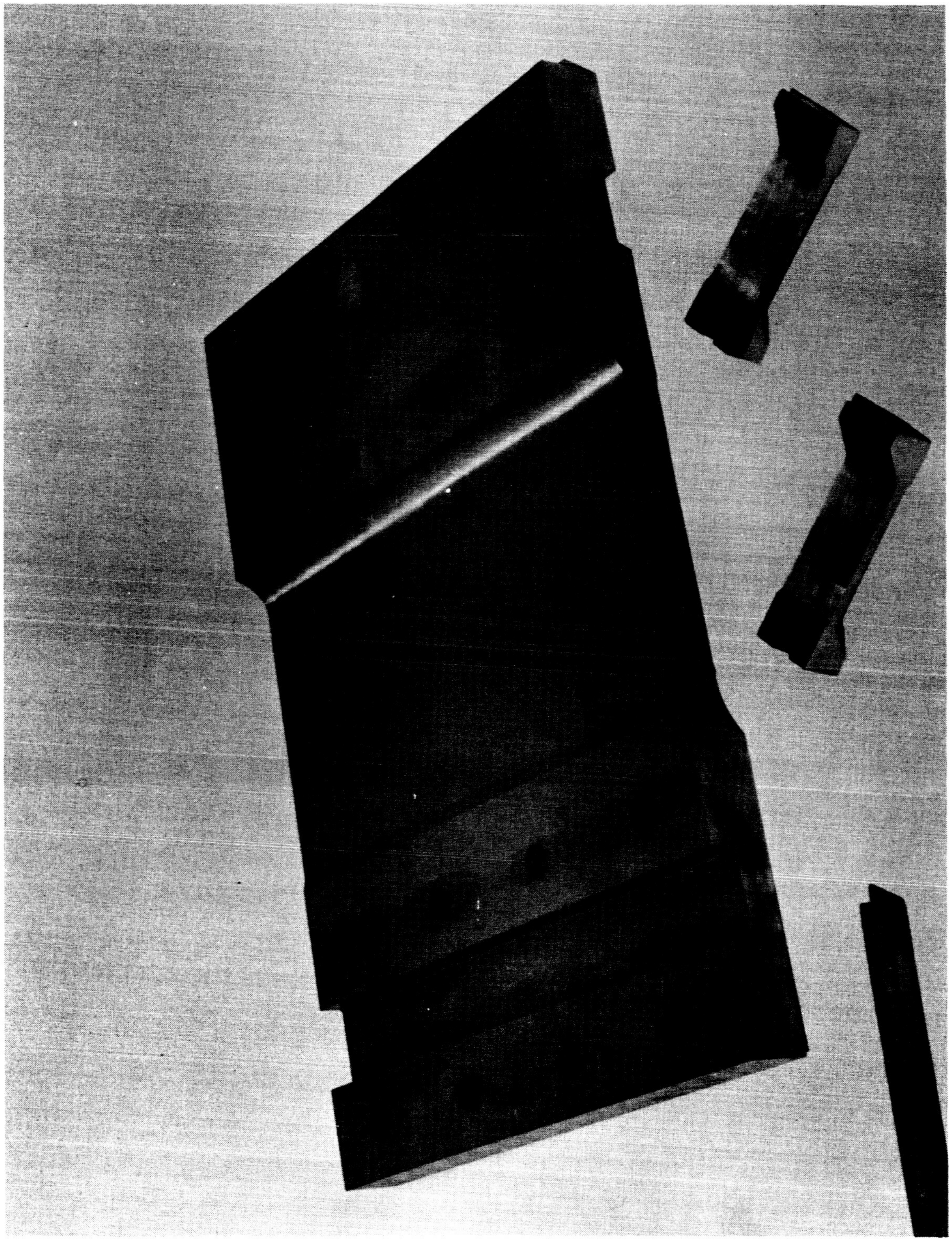


Figure 6

MAIN SUPPORT (TENSION) FLEXURE AND CROSS-TIES dead-weight load from the stud holder (21) to the pick-up beam (19), and the other two flexures carry the dead-weight load from the pick-up beam (19) to the force multiplying beam (12). The latter two flexures are mounted so that the center line of rotation is the same as the pivot (11) and loading (14) flexures when the beam is in a balanced condition. To ensure stability of the weight pick-up system, two stabilizing struts (25) with flexures at each end were installed between the weight pick-up system and the ground frame of the 60,000 lb machine.

The weight pick-up system, which is above the floor level, is hinged at the upper end of the pick-up columns (18) and can be lowered into the pit to allow unobstructed use of the 60,000 lb dead-weight machine.

5. Destabilization System

To neutralize the restoring moments of the main flexures (11), two flexure destabilization units (26) were installed below the action line of the tension flexures. Each unit is essentially a spring-loaded knife edge.

Because the center of gravity of the system is below the center of rotation and the flexure stores energy when displaced, the beam naturally tends to seek the level position. This condition creates a dead span which would be intolerable in this device. In an attempt to eliminate this condition, a destabilizing device composed of a knife edge pushing against a spring was devised (see Figure No. 2 (26) and Figure No. 7). This knife edge stores maximum energy in the spring when the beam is level. When the beam is displaced, the knife edge and spring rock past the center. The spring tends to elongate and the knife edge creates a component of force parallel to the beam. This generates a moment overcoming the combined moments of the stored energy of the flexures and the center of gravity shifting with respect to the main tension flexures.

Subsequently, this system exhibited wear on the knife edges which caused non-repeatability and hysteresis to be excessive.

To accomplish the same results, the effective center of gravity of the beam was raised above the center of rotation by raising the counter-balance weight approximately 12-in. and removing the destabilizers. Subsequent calibrations of the system show no appreciable non-linearity or hysteresis.

6. Traveling Platen (Beam)

The traveling beam of the force multiplying system has the following three functions (see Figure No. 8).

a. By moving the traveling beam either up or down, the size of the opening for the thrust cell can be controlled. For compression calibrations, the traveling beam (27) is moved to an up position and the thrust cell is placed between the lower face of the traveling beam (27) and the upper surface of the load yoke (15). Figure No. 9 shows a one-million pound capacity thrust cell installed for compression calibration. For tension calibrations, the traveling beam (27) is

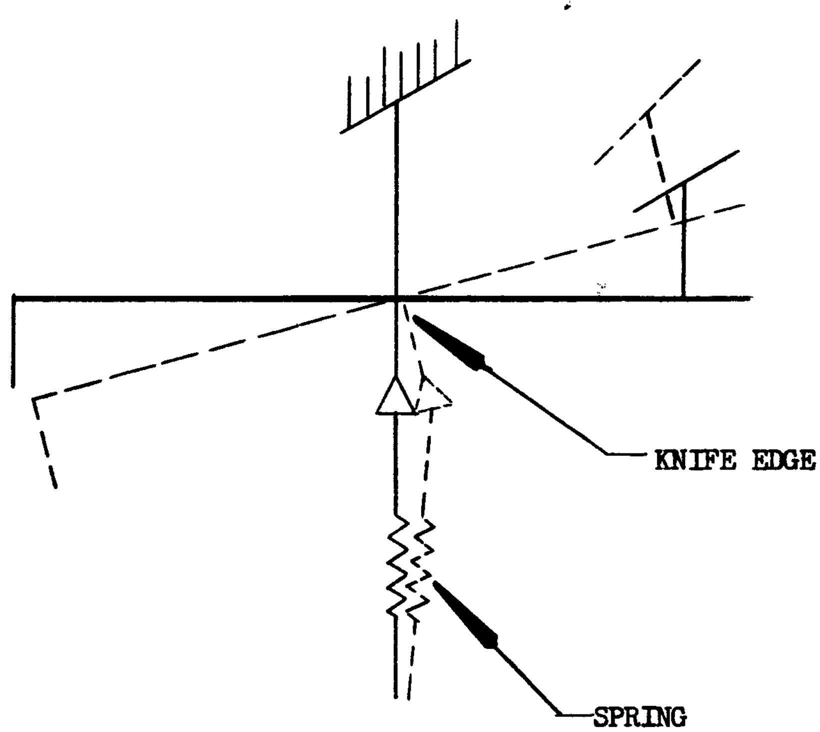


Figure 7

DESTABILIZER REACTIONS UNDER BEAM DISPLACEMENT

Figure 8

TRAVELING BEAM AND BALL SCREWS

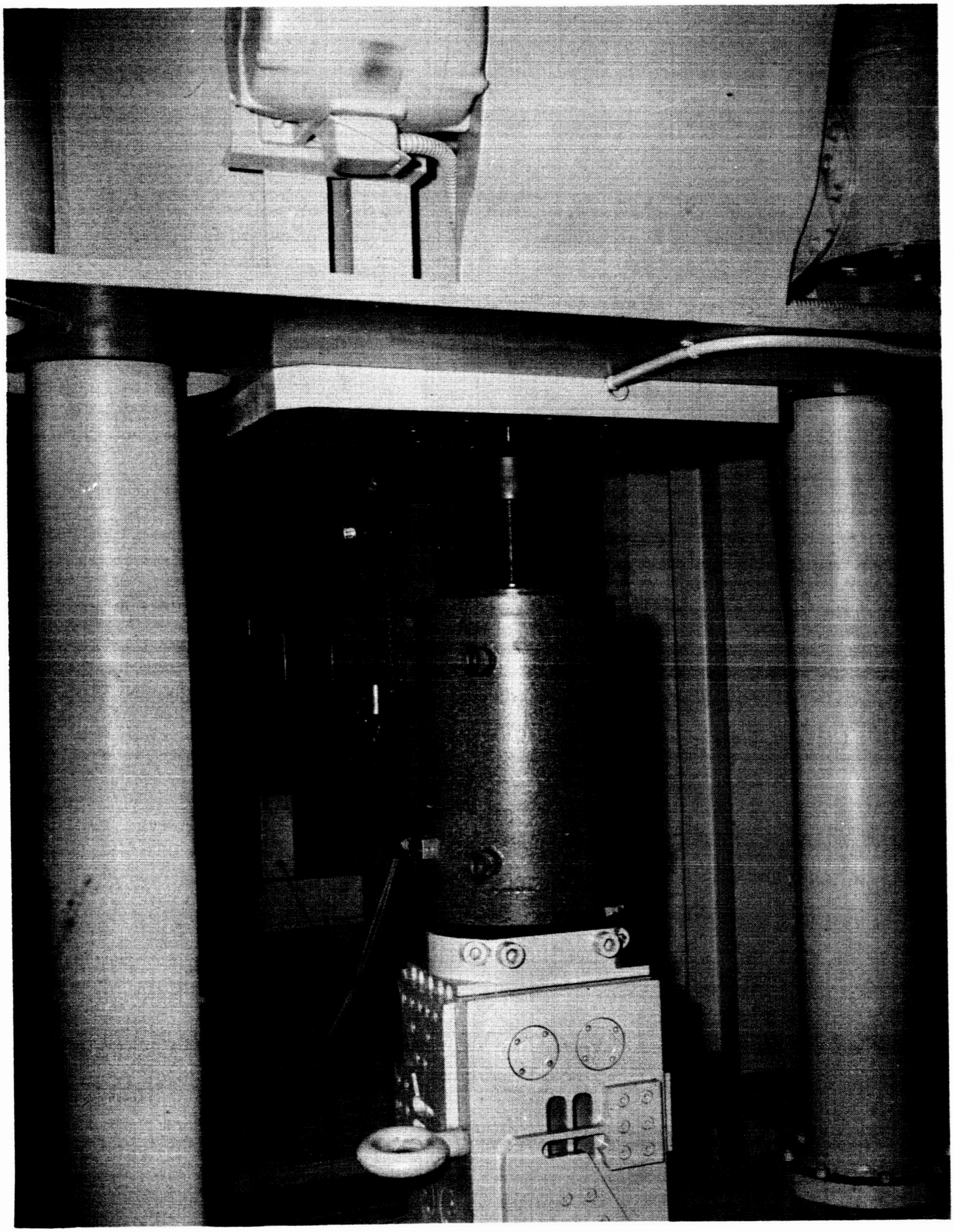


Figure 9

CELL MOUNTED FOR COMPRESSION LOADING

moved to the down position and thrust cell is mounted between the upper surface of the traveling beam (27) and the lower surface of the upper load yoke beam (28). Figure No. 10 shows a 1.5-million lb capacity thrust cell mounted for tension calibration. The maximum opening of the force multiplying system is a height of 8-ft 8-in. and a width of 4-ft. However, the full 8-ft 8-in. cannot be utilized for tension calibrations because of the requirement for adapting the studs.

b. The traveling beam also transmits the input force from the thrust cell to the four screw jacks (6) to the ground frame (1). Figure No. 8 shows the traveling beam with the ball screw jacks as an assembly.

c. The traveling beam (27) is used to re-level the force multiplying beam (12) after weight has been added or removed from the beam. The traveling beam can be moved at two speeds. The first speed or fast drive is at a rate of 4.6-in./min and is used only to position the traveling beam under a no-load condition. The second speed or slow drive is used to re-level the force multiplying beam at each load increment during a calibration. The rate of travel is .020-in./min. The drive system of the traveling beam is shown on Figure No. 11. The high-speed drive motor (A) transmits torque to the main shaft (B). The pneumatic clutch (D) is disengaged. For the slow speed drive, the motor (C) drives the main shaft through the pneumatic clutch (D) which is energized. Both systems are reversible. The main shaft (B) through the right angle, pinion gears (E) drives the four ball screw jacks (G). Two left-hand and two right-hand ball screw jacks are used to equalize the torque applied to the traveling beam.

7. <u>Leveling System</u>

An electronic level is installed at a location 5-in. in front of the main compression loading flexures and in the center of the beam (see Figure No. 12). The electronic level is a system which has a pendulum in conjunction with two variable inductance transducers. As the pendulum swings to either side of a null or zero degree position, an error signal is observed on the readout in the control panel (see Figure No. 13). The readout has three scales with the most sensitive being $\pm$ 50 sec of arc. As the beam goes out of level, the meter needle activates a microswitch which energizes the slow drive system of the traveling beam in the proper direction. As the beam approaches a level condition, the microswitch is deactivated and the slow drive system is de-energized. The resolution of the level indicator on the $\pm$ 50 sec scale is one-half second.

8. <u>Control System</u>

The panel for the control system of the force multiplying system is shown in Figure No. 13.

a. Control Panel Functions

The following functions or observations can be made from the control panel:

Figure 10

CELL MOUNTED FOR TENSION LOADING

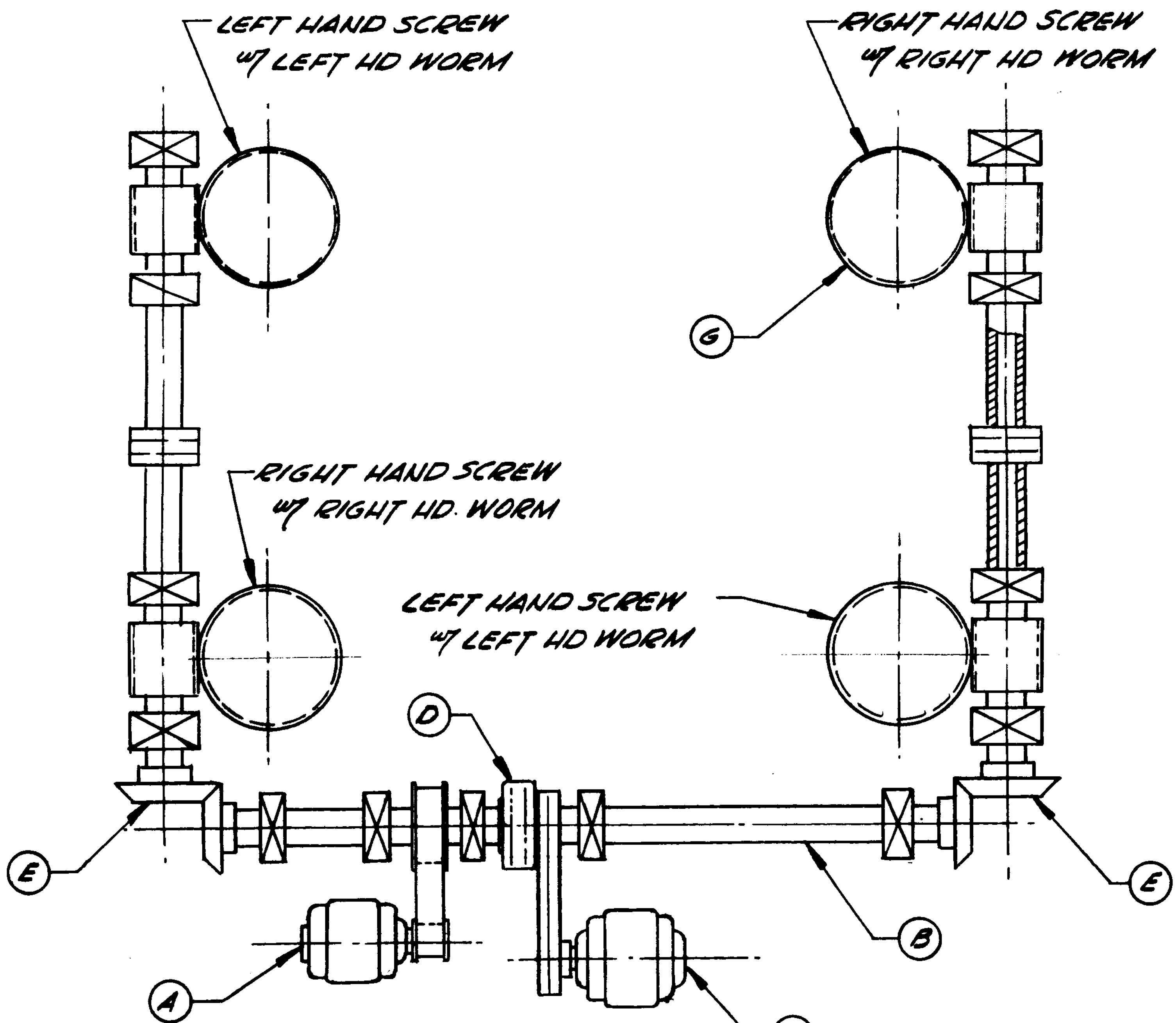


Figure 11

TRAVELING BEAM DRIVE SYSTEM

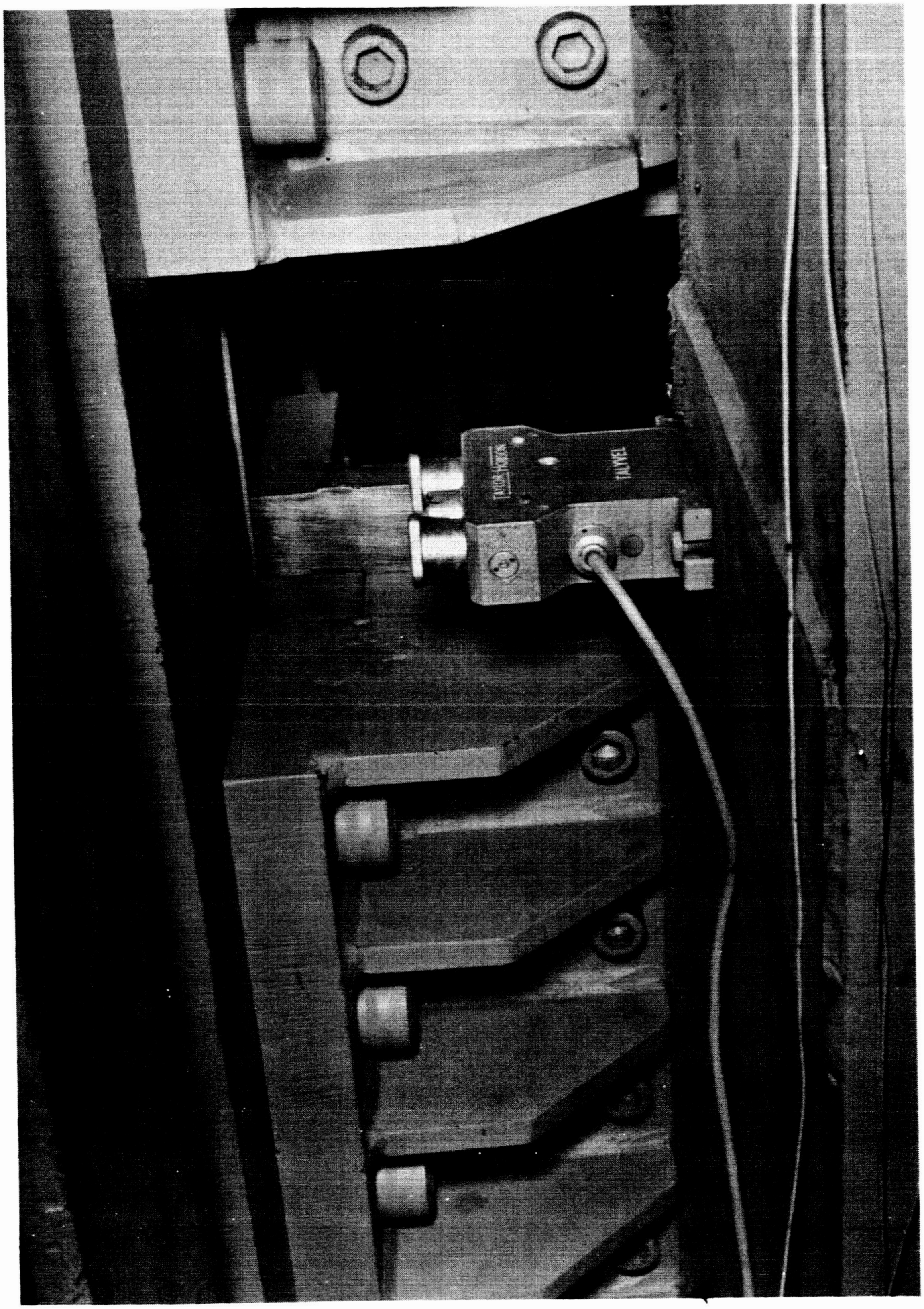


Figure 12
TALYVEL ELECTRONIC LEVEL

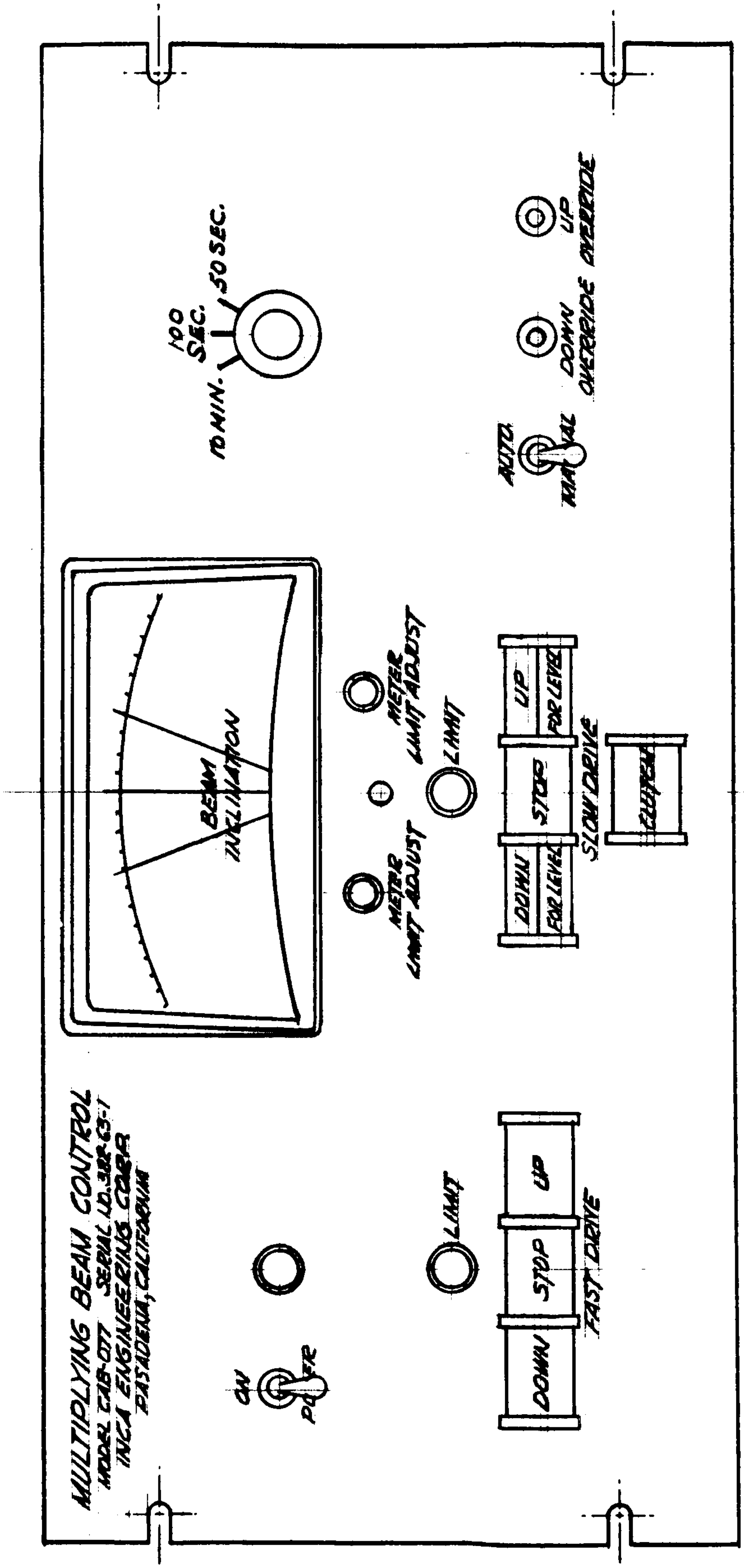


Figure 13

Control Panel

(1) Transfer of the control system from the 60,000 lb dead-weight calibrator to the force multiplying system (weight position indicator, hydraulic pump operation and valve actuation).

(2) Control of the high-speed drive motor.

(3) Control of the low-speed drive motor and clutch (manual mode).

(4) Visual indication of beam inclination.

(5) Visual indication of beam limit.

b. Switches

The switches perform the following functions (refer to Figure No. 13):

(1) <u>On-Off Switch</u>

Supplies the power to the force multiplying system and transfers the control from 60,000 lb dead-weight machine to the force multiplying system.

(2) <u>Down-Stop-Up</u>

These pushbutton switches are used for the fast drive control. They are electrically interlocked in safeguard against a sudden reversal of the motor which could cause damage. It is necessary to push the "stop" button before changing direction.

(3) <u>Down for Level - Stop-Up for Level</u>

The slow-speed drive is controlled with this set of pushbutton switches. As with the fast-speed drive control switches, an electrical interlock exists. In addition, an electrical interlock is made between the fast-speed and slow-speed drives.

(4) <u>Clutch</u>

This switch controls a solenoid valve which supplies the air for the activation of the pneumatic clutch.

(5) <u>Meter Limit Adjust</u>

The setting of these controls sets the maximum allowable beam inclination before the slow-drive system will go into operation to relevel the system.

(6) <u>Automatic-Manual</u>

With this switch in the manual position, the slow-drive system must be activated manually. In the automatic position, any time the beam inclination exceeds the pre-set limit, the slow drive system will actuate and return the beam to a level condition.

(7) <u>Up Override and Down Override</u>

If the beam inclination is within the pre-set limit and a closer beam level is desired, a momentary push of the proper override button will actuate the slow-drive system.

(8) <u>Scale Selector Switch</u>

With the selection of the three available scales, a visual indication of beam inclination can be monitored at all times.

9. <u>Weights</u>

The weights used to apply the loads to the force multiplying system are the same ones used for the 60,000 lb dead-weight calibrator. Figure No. 14 is a schematic of the 60,000 lb calibrator. The weights are stacked in a sequence which will give full scales of 5,000 lb, 10,000 lb, 20,000 lb, and 50,000 lb in five equal steps for each range. This provides for the ability to define linearity and hysteresis curves of thrust cells by incremental loading and unloading of weights. The weights are disc shaped with diameters and heights of approximately:

500 lb	21-in. by 5-in.
1,000 lb	32-in. by 5-in.
2,000 lb	41-in. by 6-in.
4,000 lb	50-in. by 8-in.
10,000 lb	70-in. by 10-in.

The construction of the weights (Figure No. 15) are such that the bottom of each weight has a ball seat (1) and the top of each weight has a ball (2). As the weights are picked up, the clearance between the ball seat and the ball of the next weight has a movement of .75-in. upward before engaging the next weight. By picking the weights up in sequence and moving the upward .75-in. for each weight up to the last weight, which is then picked up .375-in., the weights will hang free in space to give a true dead-weight load.

The control system for lifting the weights consists of a photo-sensing element and a sequencing panel. The photo-sensing system is a bar with slots spaced to coincide with each weight. The bar is attached directly to the weight stack (see Figure No. 14) (1). As each light slot passes the photo cell, a pulse is detected and a stepper switch is advanced one position. By pre-selecting the desired

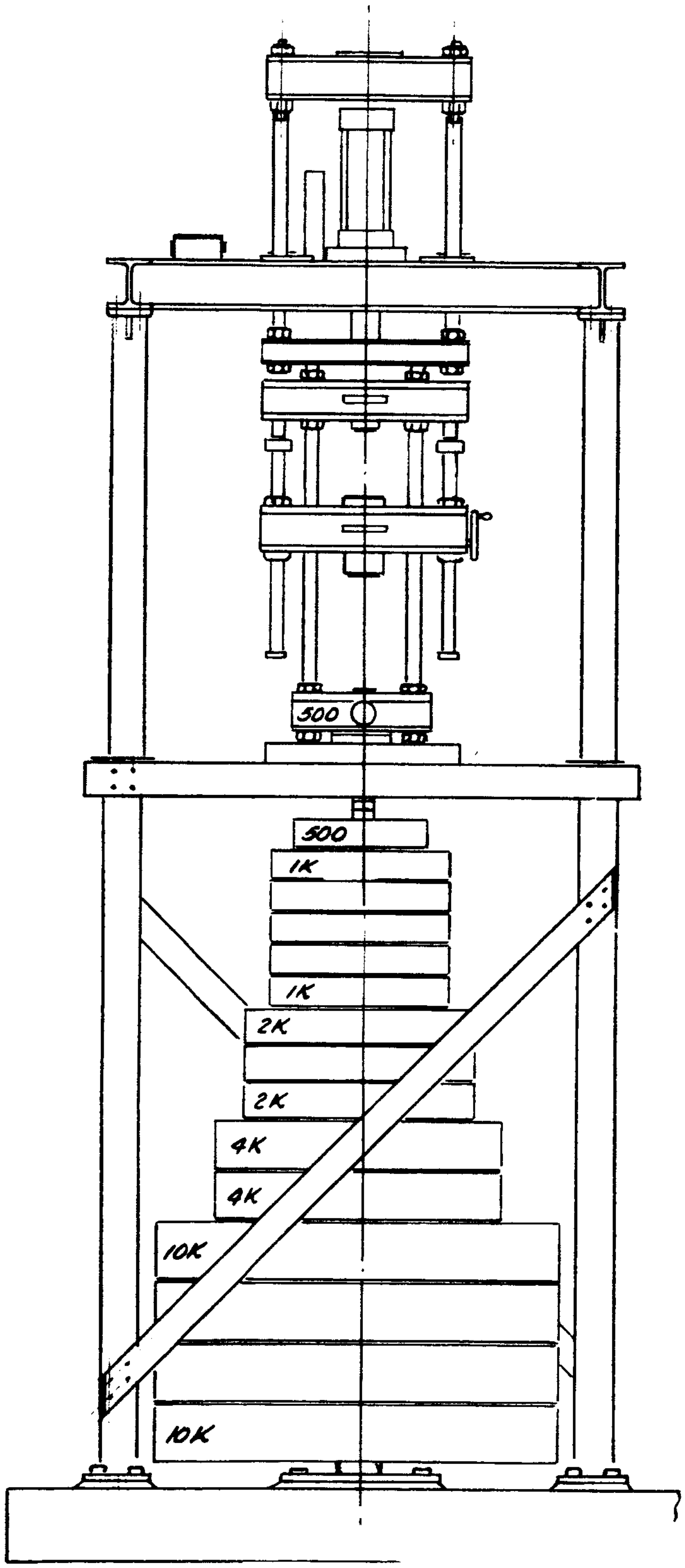


Figure 14

60,000-Pound Calibrator

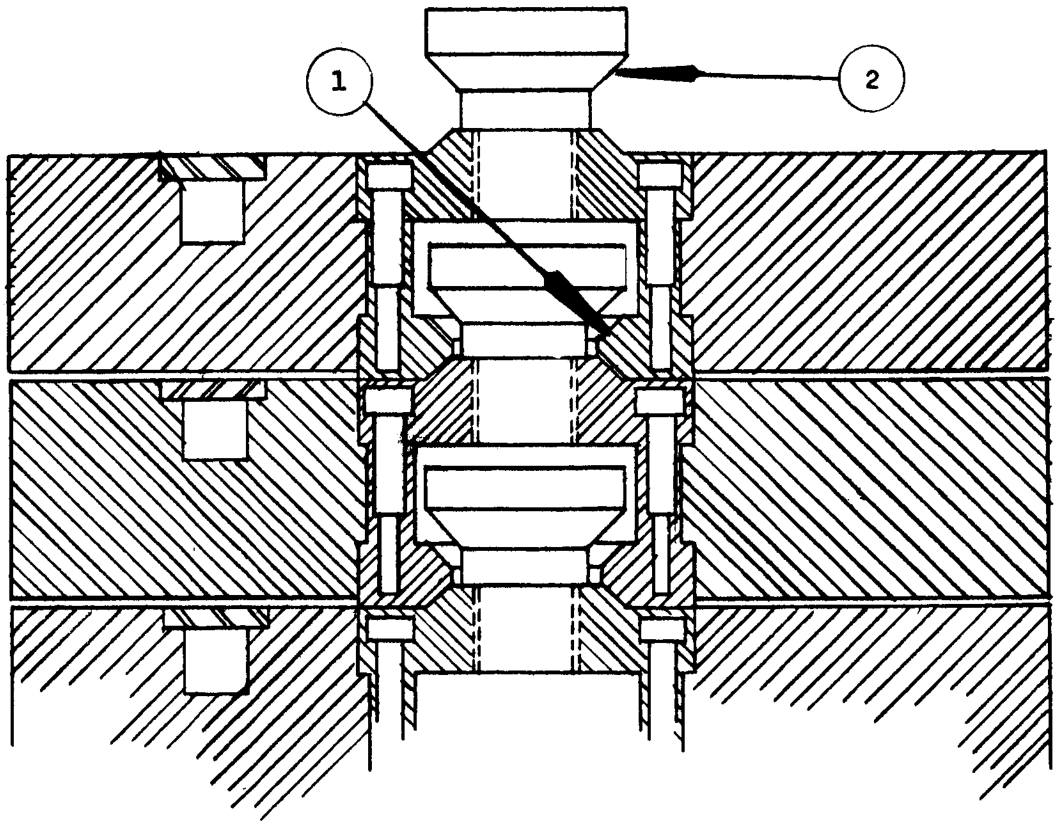


Figure 15

Cross Section of Weights

weight value on the stepper switch contacts, the system will pick up the selected weights. The same system is used for the force multiplying system, with the exception of the photo cell. When the control panel of the force multiplying system is turned on, the photo cell attached to the weight pick-up system of the force multiplying system is substituted for the photo cell on the 60,000 lb dead-weight machine (see Figure No. 16 and Figure No. 17).

The weights were calibrated and adjusted by the National Bureau of Standards to be within $\pm$ .005% of nominal value before installation. The calibration report is provided as Appendix B.

B. METHODS FOR DETERMINING BEAM RATIO

To use the force multiplying system for the calibration of thrust measuring cells, it is necessary to determine the multiplication ratio of the system at all loading points. There are four methods that could be used to determine the effective multiplication ratio of the system. These are: the direct measurement of arm lengths by optics; the use of a series of lower range cells calibrated against dead weight; the use of Morehouse proving rings calibrated by the National Bureau of Standards; and a secondary beam with the use of incremental dead weights.

1. Optical Measurement

The measurement of the effective arm ratio by the optical method was ruled out because of the construction of the force multiplying system. With the system being installed in a pit below floor level, no optical lines of sight could be established to measure the length of the short (six inch) arm.

2. Multi-Cell Determination

This method of ratio determination incorporated the use of twelve load cells. This set of cells consisted of four cells with full-scale ranges of 50,000 lb and eight cells with full-scale ranges of 200,000 lb. Each 50,000 lb cell was calibrated on the 60,000 lb dead-weight calibrator in order to have a reference to dead-weight on the lower range load cells. To ensure repeatability and achieve the best possible accuracy, each cell was rotated through 360 degrees in increments of 90 degrees. This method showed that there were negligible errors resulting from bending moments in the load cell. Figure No. 18 shows a 50,000 lb cell mounted for this calibration step.

After the 50,000 lb load cells were calibrated, each 200,000 lb load cell was then calibrated against the 50,000 lb load cells. This was accomplished by placing the four 50,000 lb cells in parallel in the force multiplying system and then placing the 200,000 lb cell in series with this combination. Figure No. 19 shows the cells in place. Again, each 200,000 lb cell was rotated through 360 degrees to determine any bending moment errors. Again, these errors were negligible.

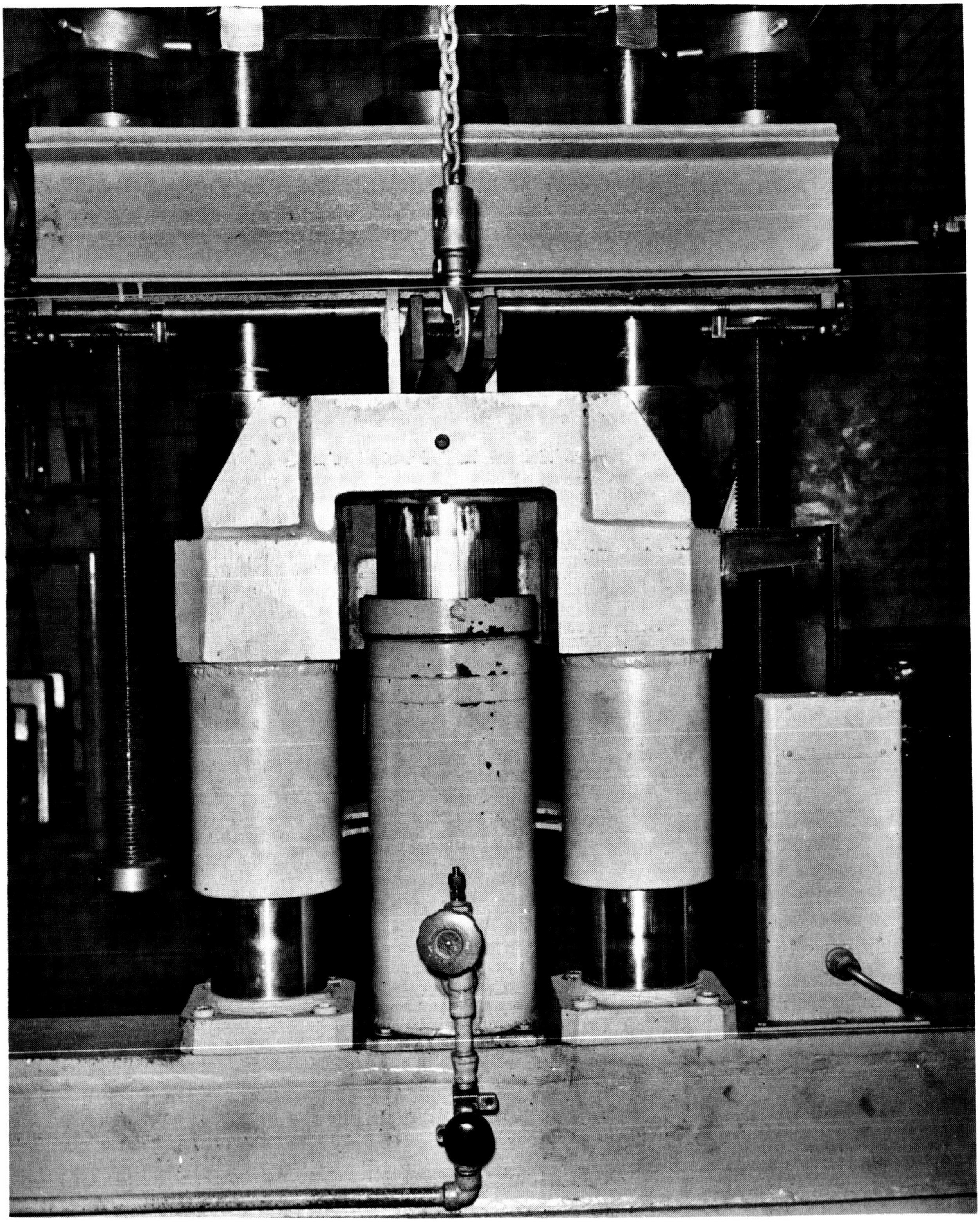


Figure 16

HYDRAULIC RAM

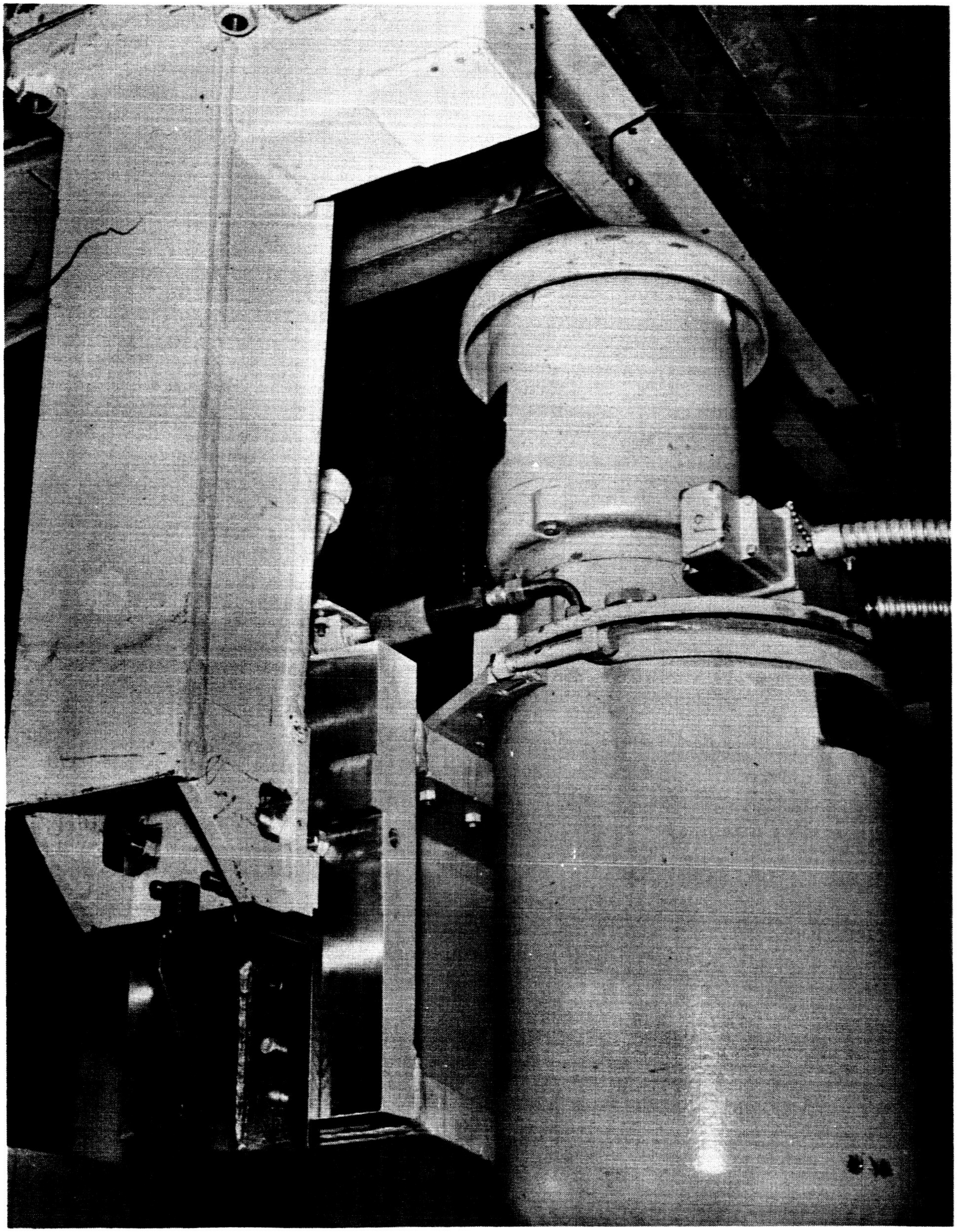


Figure 17

HYDRAULIC PUMP

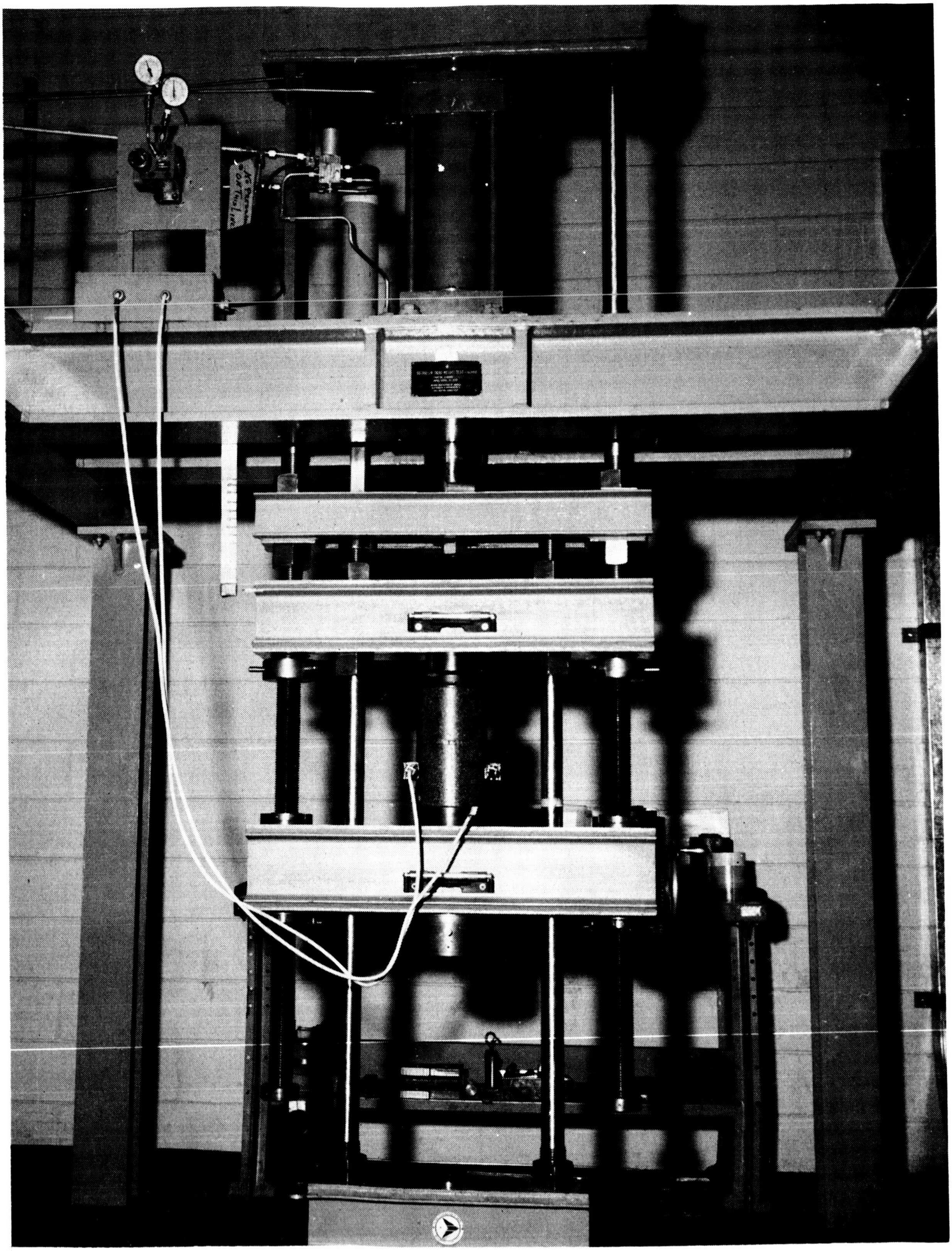
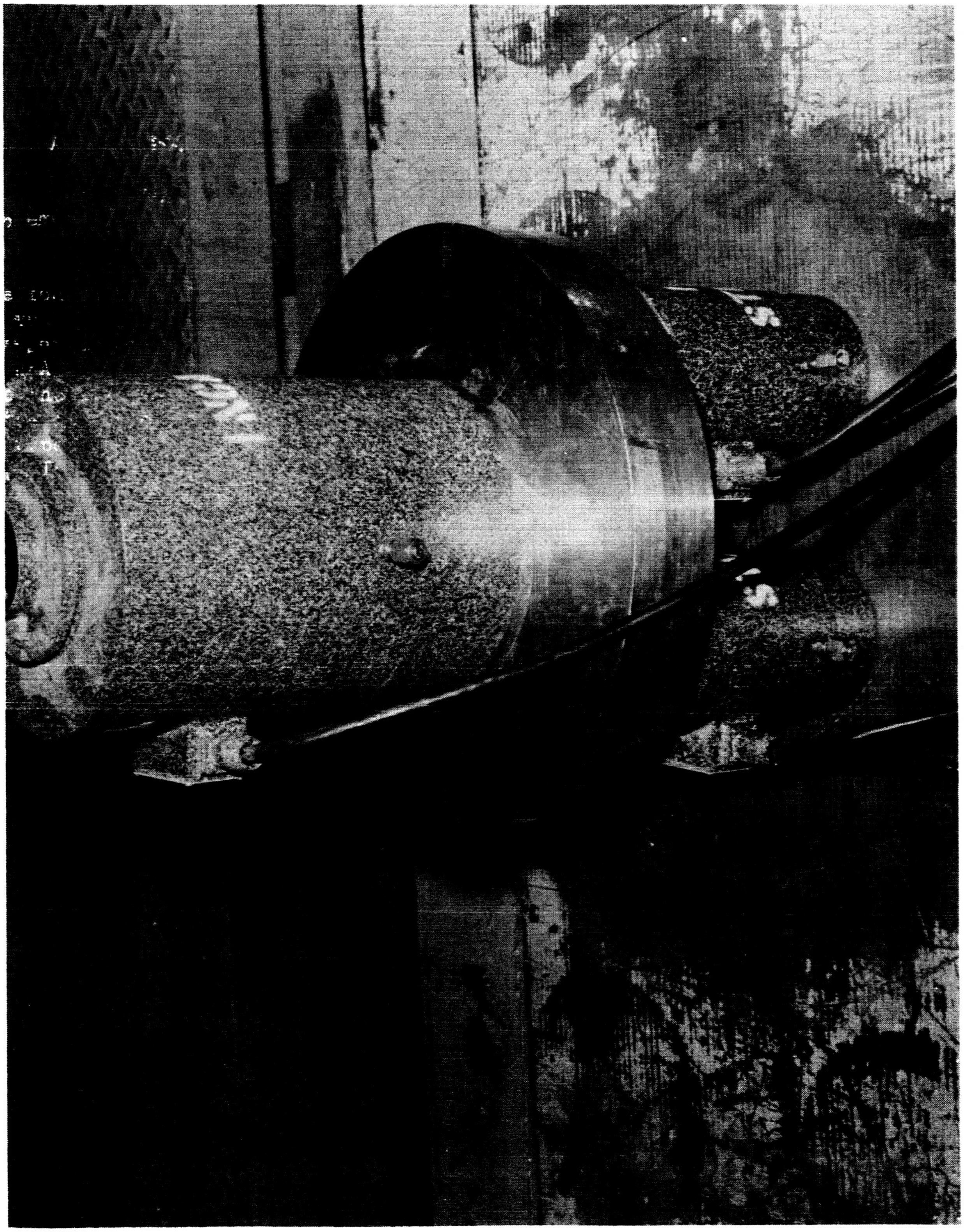


Figure 18
50,000-POUND CELL BEING CALIBRATED



200,000-POUND CELL CALIBRATION

Figure 19

After completing the cell calibrations, each range of the force multiplying system was then calibrated in the following cell combinations. The 125,000 lb full-scale range was calibrated against three 50,000 lb cells in parallel. The 250,000 lb full-scale range was calibrated with two 200,000 lb cells in parallel. The 500,000 lb range was calibrated against three 200,000 lb cells and the 1,500,000 lb range was calibrated against eight 200,000 lb cells. The final results from this type of calibration are shown in Appendix C. Results obtained in the final calibration using this technique do not appear to be indicative of the machine. An analysis of these cells indicated unequal loading of up to 15%. The National Bureau of Standards had taken extreme precautions to load rings equally within one percent. Generally, the reason given for this precaution is that rings are susceptible to bending moments and non-uniform loading results in this condition. Because of this, the calibrations were considered invalid.

3. Determining Beam Ratio with Morehouse Proving Rings

Because the results of the multi-cell calibration were not satisfactory, a second calibration was made against a series of Morehouse proving ring force standards that had been calibrated by the National Bureau of Standards. In addition to the proving rings, one load cell was used to calibrate the lower load points of the system.

The lower two load points were calibrated against a load cell which, in turn, had been calibrated directly against the dead weights.

The load points between 75,000 lb and 1,250,000 lb were calibrated against the proving rings. Each proving ring was load-cycled to full scale until the zero load point became repeatable.

Because Circular No. 454 of the National Bureau of Standards sets the limits of the proving rings as being "one-tenth of one percent of the deflection for the capacity load" only the upper load points of each ring were used. The results of this calibration are shown in Appendix D. The average beam ratio was 25.0075:1. From this data, it can be concluded that the force multiplying system is accurate to within one-tenth of one percent throughout the full range of the system.

4. Secondary Beam

In an attempt to improve the calibration of the force multiplying system by the use of incremental dead weight, a secondary beam was used. To date, no success has been achieved with this type of calibration.

IV. PROBLEM AREAS

The following problems were encountered with the force multiplying system. These problems are of the nature which cannot be predicted in advance of the system check-out.

A. CROSS-TIE FLEXURES

There are two locations of cross-tie flexures that have caused some problems.

The cross-ties of the main loading flexures, shown on Figure No. 20, were found to have some lateral movement. This lateral motion caused the multiplying arm to shift and reflected a change; therefore, non-repeatability to exist. An investigation of the problem revealed that the stabilizing struts (Figure No. 2, Callout 25A) were not parallel with the multiplying arm. A force was being transmitted into the auxiliary support beams (Figure No. 2, Callout 5). A bending motion caused by the length of the auxiliary support beams was observed. This condition allowed the upper load yoke beam to rotate and induce a lateral movement into the multiplying arm. The ends of the stabilizing struts were then moved to a beam of the building and set parallel with the multiplying beam. After this was accomplished, the lateral motion observed in the multiplying beam was of insignificant magnitude.

One of the weight pick-up cross-tie flexures was broken (Figure No. 2, Callout 24) during the lowering process of the weight pick-up system (Figure No. 2, Callout 21). An investigation revealed that the weight of the weight pick-up (Figure No. 21) system must be fully supported at all times when the system is being lowered to the out-of-use position.

B. ECCENTRIC LOADING OF THE MAIN FLEXURES

Currently, the main loading flexures are being loaded eccentrically. Figure No. 22 is a cross-section of the interface of the loading flexures and the multiplying arm. The input force (A) is transmitted through the plate (4). The force is transmitted through the flexure key (1) to the flexure (2). The cross-tie flexure is callout (3). The force is then transmitted to the load yoke along the line (B). Because the force lines (A) and (B) are not in line, an eccentric force is generated by the moment arm of (A) to (B). This moment causes the plate (4) to bend and move in the direction indicated by (C).

By installing a back-up plate (5), the distance between the load lines (A) and (B) could be reduced to a negligible factor.

C. TENSION MOUNTING ADAPTERS

During the calibration of a 1.5 million-pound load cell for tension use, the cell became jammed on the adapter stud. To remove the cell, the adapter had to be cut. Because it appeared that this could be a continuing problem, a new set of adapting hardware was designed that incorporates the use of eye rods, a clevis, and clevis pins. This will allow the eye rod to be installed in the load cell before it is mounted in the force multiplying machine.

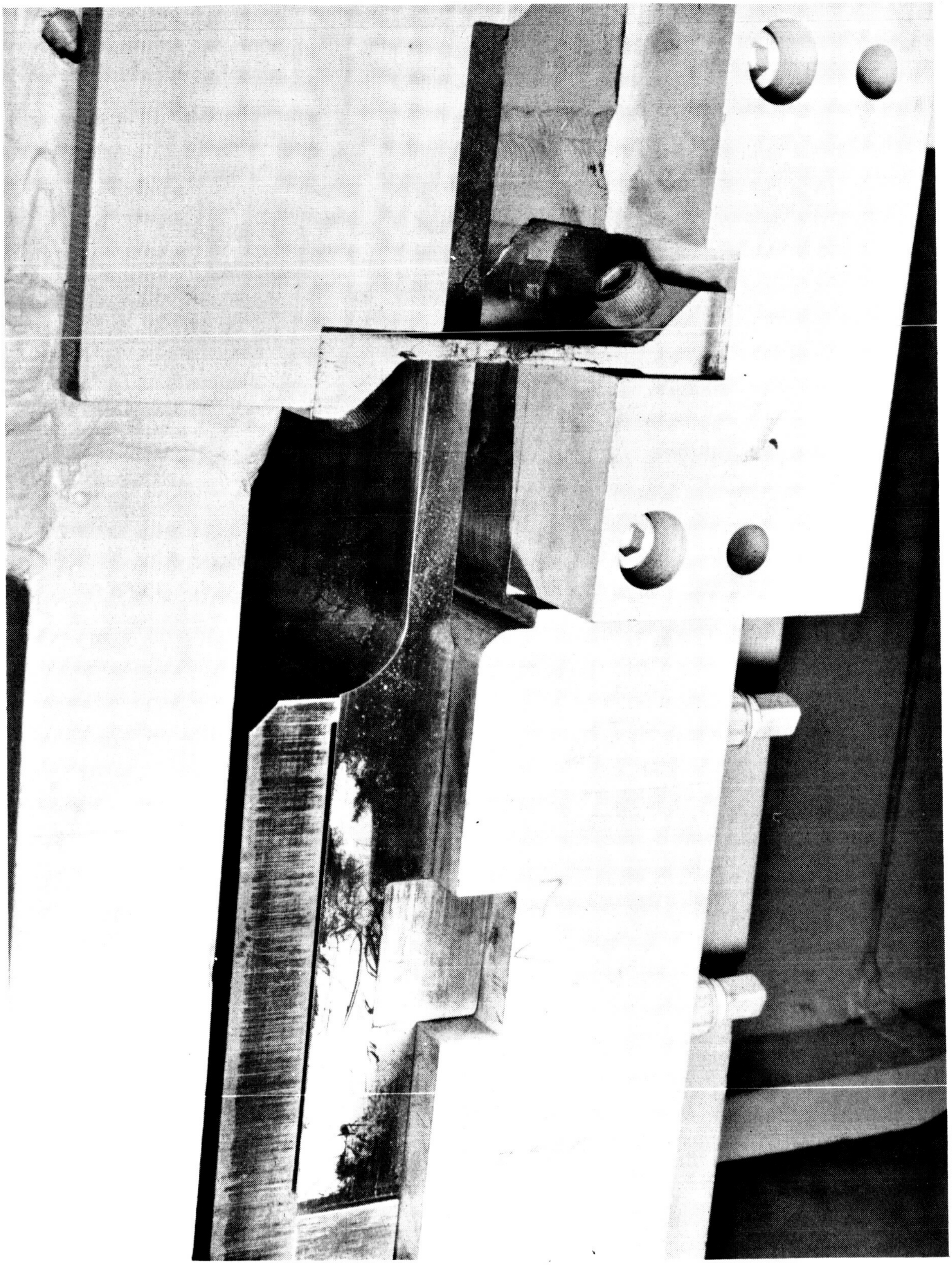


Figure 20
CROSS-TIE FLEXURE (MAIN)

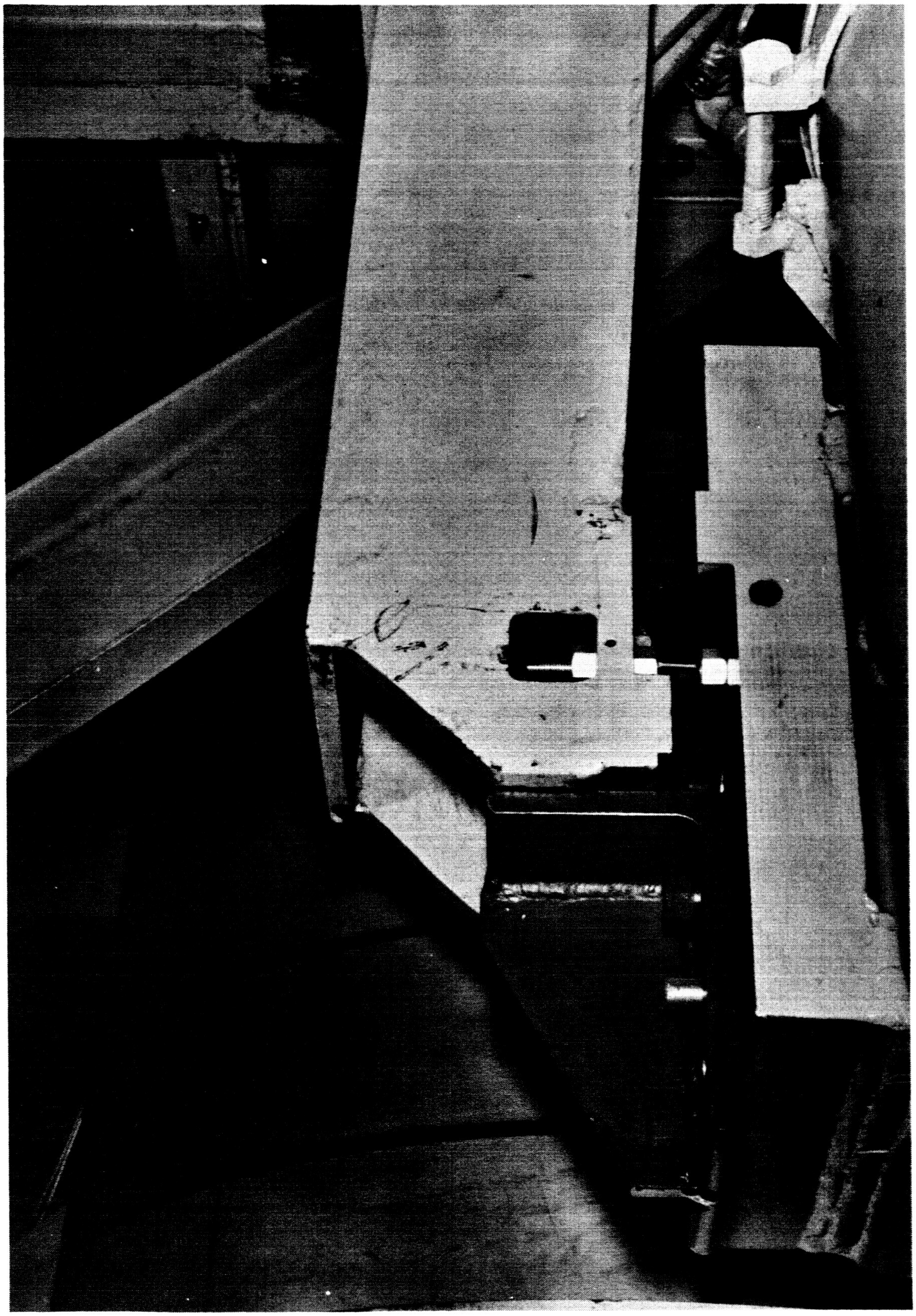


Figure 21

CROSS-TIE FLEXURE WEIGHT PICK-UP

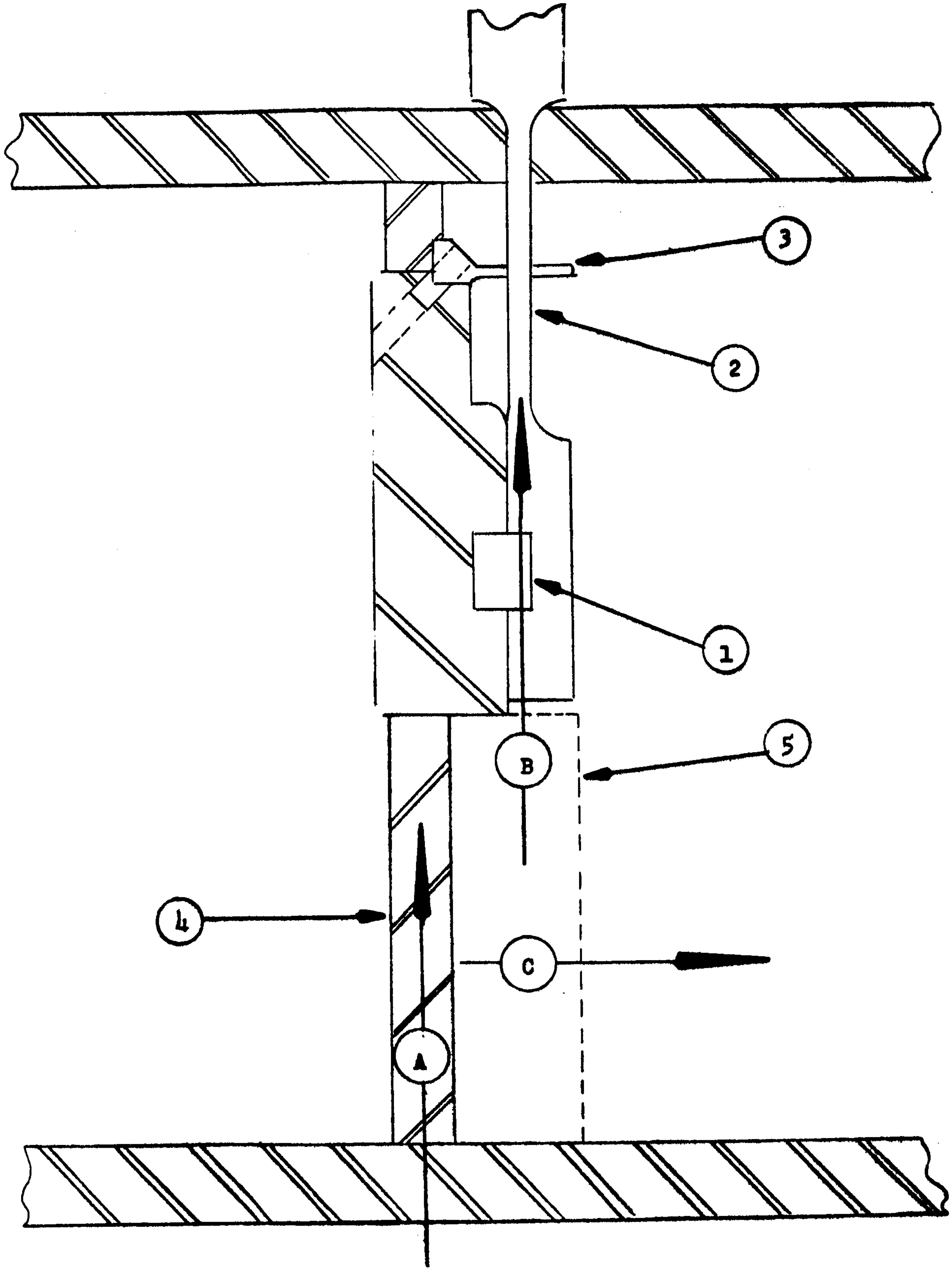


Figure 22

LOADING FLEXURES

V. RESULTS OF THRUST CELL CALIBRATIONS

The results of the calibration of one 1.5 million-pound capacity thrust cell are presented on Figures No. 23 and No. 24. The curve marked AGC shows the results obtained from the loading on the force multiplying system. The BLH curve is plotted from the data supplied by the manufacturer.

It is necessary to discuss the method of calibration used by the manufacturer. The thrust cell was calibrated using two different reference standards. The dotted line on the figures is the cross-over point on the reference standards. The calibration made by the manufacturer was within specification. Details of the equipment used by the manufacturer are provided in Appendix E.

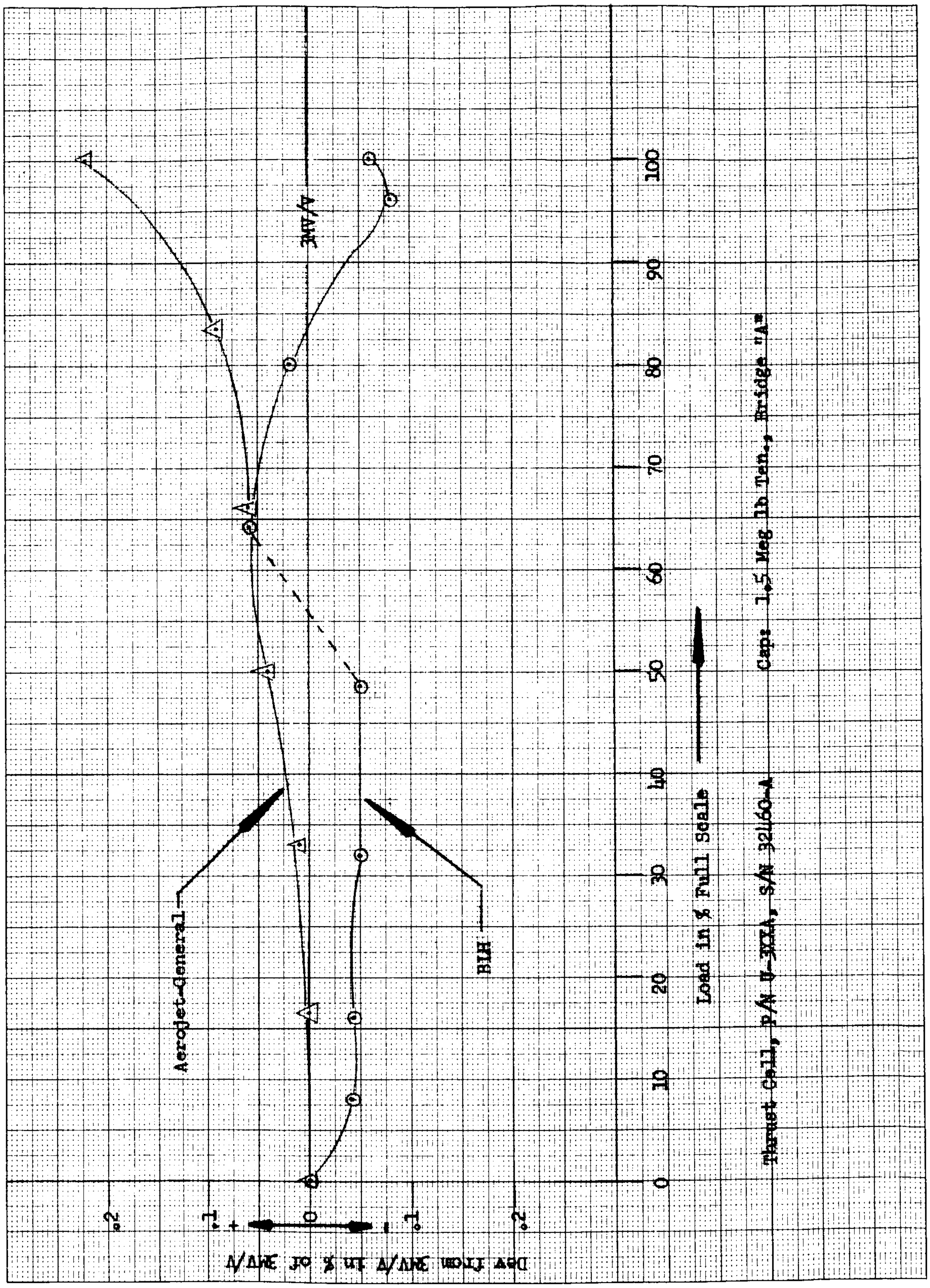


Figure 23

Calibration of Load Cell Results

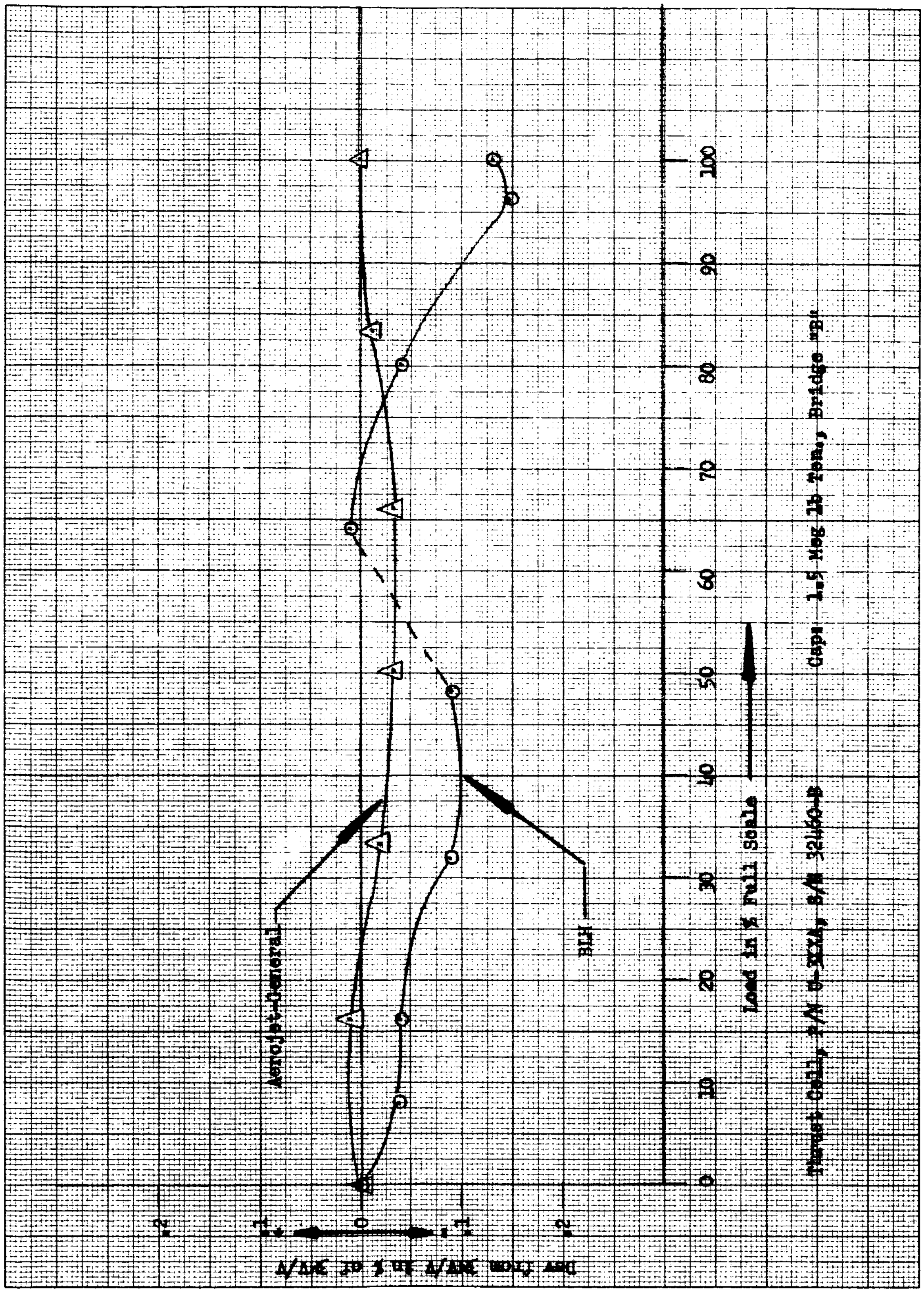


Figure 24

Calibration of Load Cell Results

PART II

H-AREA THRUST MEASURING SYSTEM

I. INTRODUCTION

The function of Test Stand H-8, located at the Liquid Rocket Test Facility, Aerojet-General Corporation, Sacramento, California, is to provide adequate data for the evaluation of M-1 thrust chamber assembly performance when test fired at sea-level conditions. The thrust measurement system located in Test Stand H-8 must provide accurate thrust chamber assembly axial thrust data which, in conjunction with propellant flow rate data and thrust chamber pressure data, permits valid evaluation of thrust chamber injector performance. The design concept for the thrust measurement system and its integral calibration system incorporates Aerojet-General Corporation rocket test experience gained through the design and operation of similar systems as well as incorporating several innovations. Because of size and thrust rating of the system, the innovations are specifically aimed at improving system repeatability, minimizing stand hysteresis, and facilitating a valid system calibration.

The major design objectives are delineated herein along with the resulting design configuration. Results of the analytical studies which support the validity of the final design configuration are also discussed. The calibration procedure is described in detail and the use of the calibration data for the purpose of reducing thrust data from thrust chamber assembly firings is explained.

II. DESIGN CRITERIA

A. DESIGN OBJECTIVE

The basic design objectives of the thrust system was to provide an axial force measurement system with maximum accuracy and repeatability. It had to be structurally capable of supporting the M-1 thrust chamber assembly and of reacting unpredicted non-axial or axial thrust loads with a reasonable design load safety factor. The basic design objective of the static calibration system was to determine the system resistance to deflection and to determine the repeatability of this resistance to motion. As a matter of practical concern, the size of the thrust system and its several components required that ease of handling, installation, adjustment, and operation of the system or its components be a prime design objective.

B. FUNDAMENTAL CRITERIA

The basic design criteria were specified in the M-1 Model Specification, which required that the thrust measurement system at Test Stand H-8 be capable of determining steady-state axial thrust of the M-1 thrust chamber assembly fired at sea-level conditions with an accuracy of $\pm$ 0.75% 3σ. Initially, the thrust/time curve was assumed to be identical to the predicted M-1 engine start transient and no non-axial forces were predicted. The start transient criterion was modified to include a hard start (Figure No. 25) based upon the results of a liquid oxygen/liquid hydrogen thrust chamber assembly computer study. This study included consideration of the effect of partially pre-pressurized propellant vessels and hydraulically-operated globe valves as thrust chamber valves as well as the characteristic of the single, full-scale thrust chamber assembly test previously performed at Test Stand C-9. Based upon a summary report supplied by the Rocketdyne Corporation of data obtained from test firings of the J-2 liquid hydrogen/liquid oxygen rocket engine, a non-axial thrust load (see Figure No. 26) was incorporated into the thrust system criteria. The J-2 data was arbitrarily scaled to M-1 proportions utilizing the ratios of thrust and thrust chamber expansion ratio.

C. CRITERIA ESTABLISHED BY EXISTING CONDITIONS

Test Stand H-8 was an existing facility originally designed and used to test fire the first-stage Minuteman motor in a horizontal attitude at a thrust level of 160,000 lb. The original thrust block was a Z-shaped, monolithic reinforced concrete structure, which was modified to accommodate the higher thrust level of the M-1 thrust chamber assembly by adding a massive pour of reinforced concrete behind the original thrust block. Inasmuch as the facility was specifically designed to accommodate a horizontal thrust load, the M-1 thrust chamber assembly system was specified to be a horizontal system and no consideration was given to reorienting the thrust centerline for the M-1 thrust system.

D. REVIEW OF APPLICABLE EXPERIENCE

Prior to selecting a final design configuration for the H-8 thrust system, an extensive review was made of applicable Aerojet-General Corporation experience. This experience included design, operation, and calibration of thrust systems for both liquid and solid rockets that were similar to the M-1 thrust system. Test stands built for the Titan Program incorporated both horizontal and vertical thrust systems utilizing single or multiple cells operating in either tension or compression. The Solid Rocket Test Facilities had extensive experience with a variety of horizontal thrust systems utilizing single and multiple load cells operating in compression as well as with plate flexure support systems and guided roller support systems. In both the liquid and solid rocket thrust measurement systems, several different types of devices had been used to isolate the thrust cell(s) from bending and shear loads. These bending and shear loads occur as a result of the relative motion of the opposing mounting surfaces of the thrust measurement system which is caused by the elastic deformation of the stand subjected to thrust load or differential thermal deflections. The several devices include a single rod flexure located on one side of the force measurement load cell, or two rod flexures located on either side of the load cell, or alternatively, two monoballs or two web flexure pivots located in either side of the load cell.

In summary, Aerojet-General Corporation experience indicated the following:

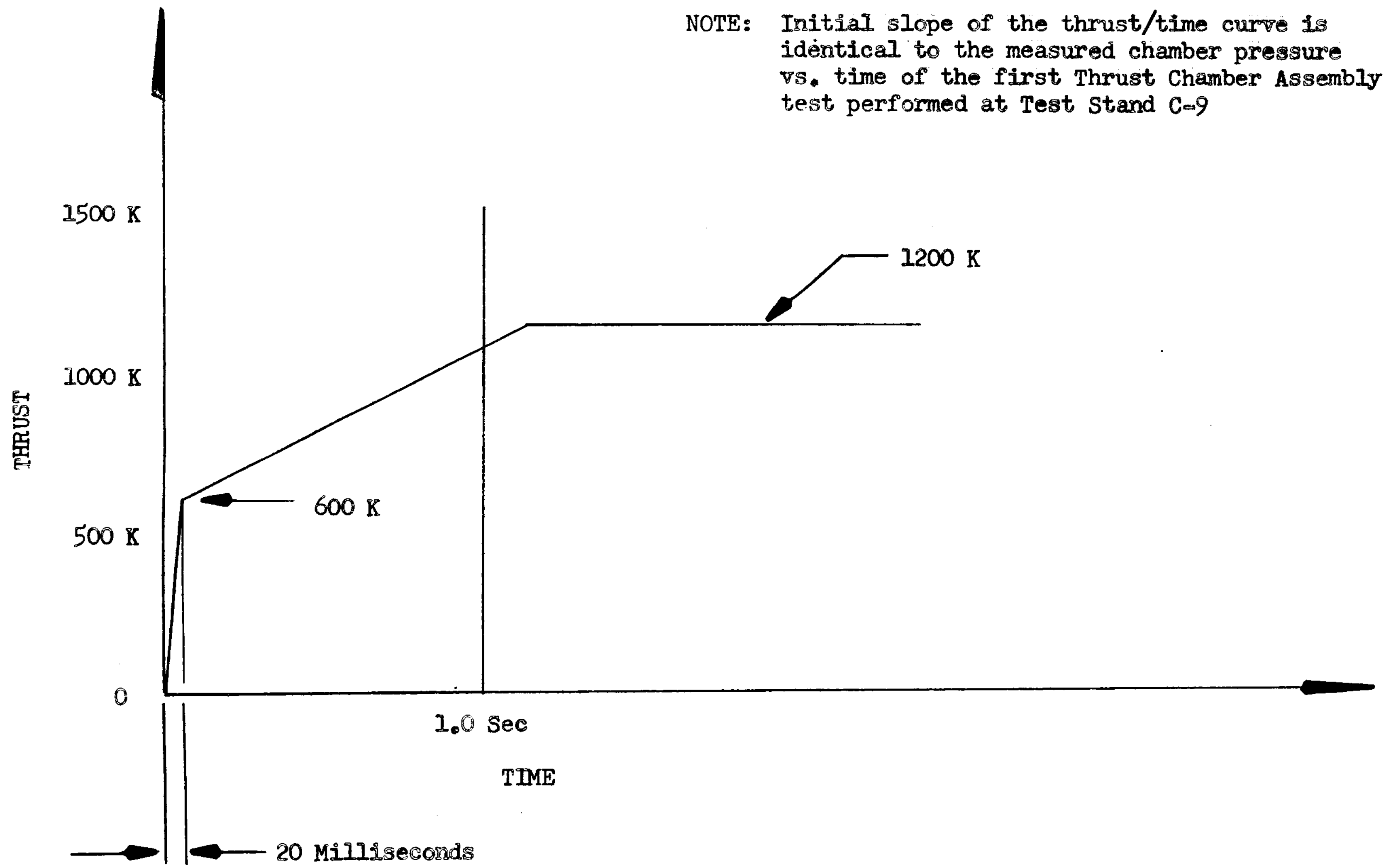
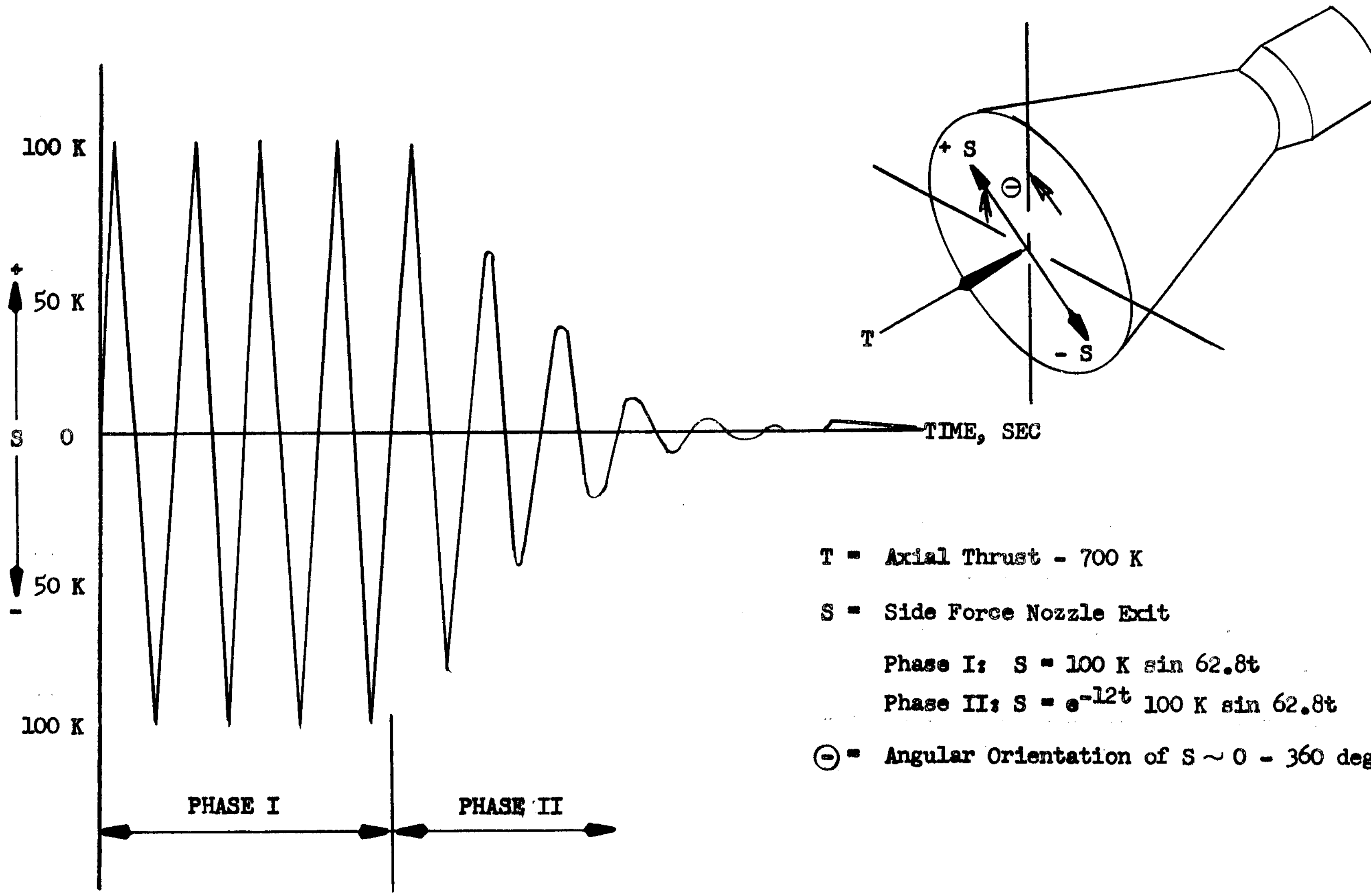


Figure 25

M-1 THRUST CHAMBER ASSEMBLY "HARD START" TRANSIENT



T = Axial Thrust - 700 K

S = Side Force Nozzle Exit

Phase I: $S = 100\ K \sin 62.8t$

Phase II: $S = e^{-12t}\ 100\ K \sin 62.8t$

⊖ = Angular Orientation of $S \sim 0 - 360$ degrees

Figure 26

NON-AXIAL LOADING RESULTING FROM ASYMMETRICAL GAS EXPANSION

1. The multiple load cells operating in tension or compression did not produce as consistent results, from system to system, as single load cell systems. This error apparently results from nonsymmetrical deflection of the structural members which join the load cells. The phenomenon was very pronounced at a Titan I second-stage vertical test stand with a thrust rating of 450,000 lb and a horizontal thrust system used to test fire the first 100-inch diameter segment solid rocket at a thrust level of 600,000 lb. A different Titan I second-stage vertical stand with a single cell in tension had an error of less than .25%.

2. Roller supported thrust systems exhibited non-repeatability in excess of ± 5%.

3. Flexure supported thrust systems exhibited non-repeatability less than ± .5%.

4. Problems associated with data reduction of transient thrust data obtained during the start transient of solid rocket test firings indicated that the thrust measurement systems accuracy of the transient force measurement is improved with increasing natural frequency of the thrust measurement system. In liquid rocket testing, high stand natural frequency has the added advantage of minimum hardware movement and consequently, minimum deflection of propellant lines and associated piping.

5. Gimbal joints installed in the propellant line to accommodate the stand motion are a source of non-repeatable restraint.

E. DESIGN CRITERIA AND CONFIGURATION

The initial design criteria and configuration selected for the thrust measurement system were:

1. A horizontal attitude.

2. The thrust system to operate in tension.

3. The structural support of the system to be plate flexures.

4. A single thrust measurement load.

5. A single calibration load cell.

6. Both the calibration and the thrust measurement cell to be isolated by web type flexure pivots placed on either side of the load cell.

7. The calibration system to be independently supported.

8. The thrust system to be designed for the highest natural frequency practical.

9. The system and support structure to be designed to react 20% of 1,500,000 lb supplied in the plane of the thrust chamber assembly mount.

10. The calibration system to be integrally mounted and capable of remote operation.

11. Handling fixture to be supplied for all major components and subassemblies.

12. The thrust system and critical surface squareness to be accurately controlled.

The above basic criteria resulted in the current M-1 thrust chamber assembly thrust measurement system with few exceptions. The initial concept is shown in Figure No. 27 and a detailed description follows. Figure No. 28 and Figure No. 29 are photographs of the completed system.

III. BASIC FACILITY DESCRIPTION

A. BASIC STRUCTURE

The M-1 thrust chamber assembly thrust system consists of a basic thrust frame supported at either end of plate flexures. The frame, which is approximately 40 ft long, is constructed of two pipe columns and a cross-beam at either end. The pipe columns have a 24-in. outside diameter and a one and one-half inch wall. The cross-beams are heavy weldments of two inch and three inch plate. The centerline elevation of the frame, which is also the centerline elevation of the frame, which is also the centerline of thrust, is 12 ft 3-in. above the deck. The two plate flexures support the weight of the frame in compression.

From the aspect of repeatability, it would have been desirable to have oriented the support flexures so that they supported the weight of the thrust system in tension. However, the tension flexure concept would have required considerably more structure to have reversed the position of ground and would have increased the cost of the system excessively.

The fore and aft support flexures are fabricated of plate. T-beam stiffeners are welded to either side of the plate to prevent the central portion of the flexure from buckling. The top and bottom edges are not reinforced with stiffeners. This unreinforced portion of the plate becomes the pivot or hinge line of the flexure. The fore and aft flexures are not identical because each flexure has been specifically designed to react the weight and non-axial thrust component at that station. The forward flexure which is closest to the thrust wall, is a reinforced half-inch plate that is eight feet wide and approximately 10 ft high. It is necessary that the base of the aft flexure be 15 ft wide and one inch thick to accomodate the high shear and moment loads at this station. The top portion of this flexure is eight inches wide and one and one-half inches thick.

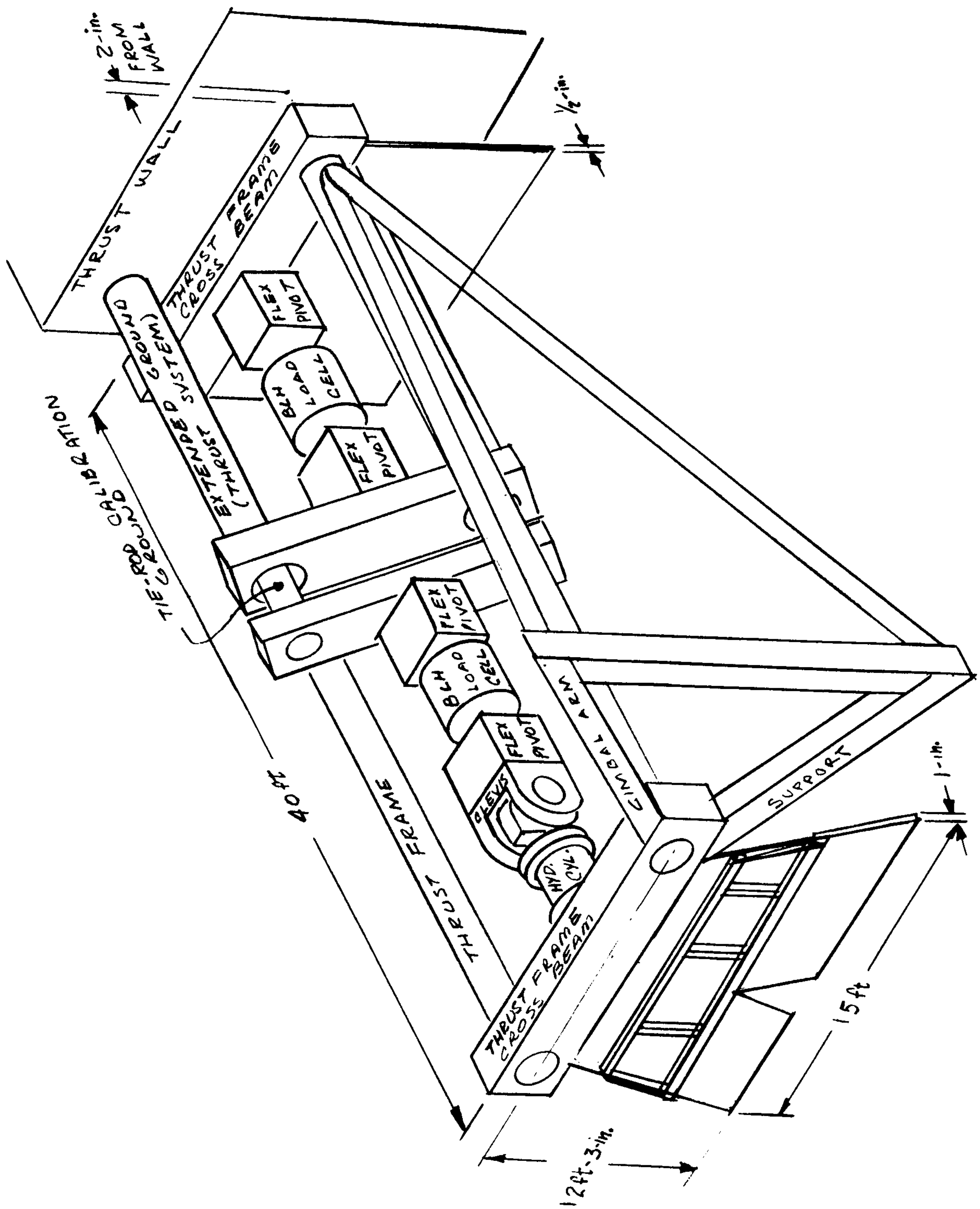


Figure 27

M-1 TCA Thrust Measuring System, Original Configuration

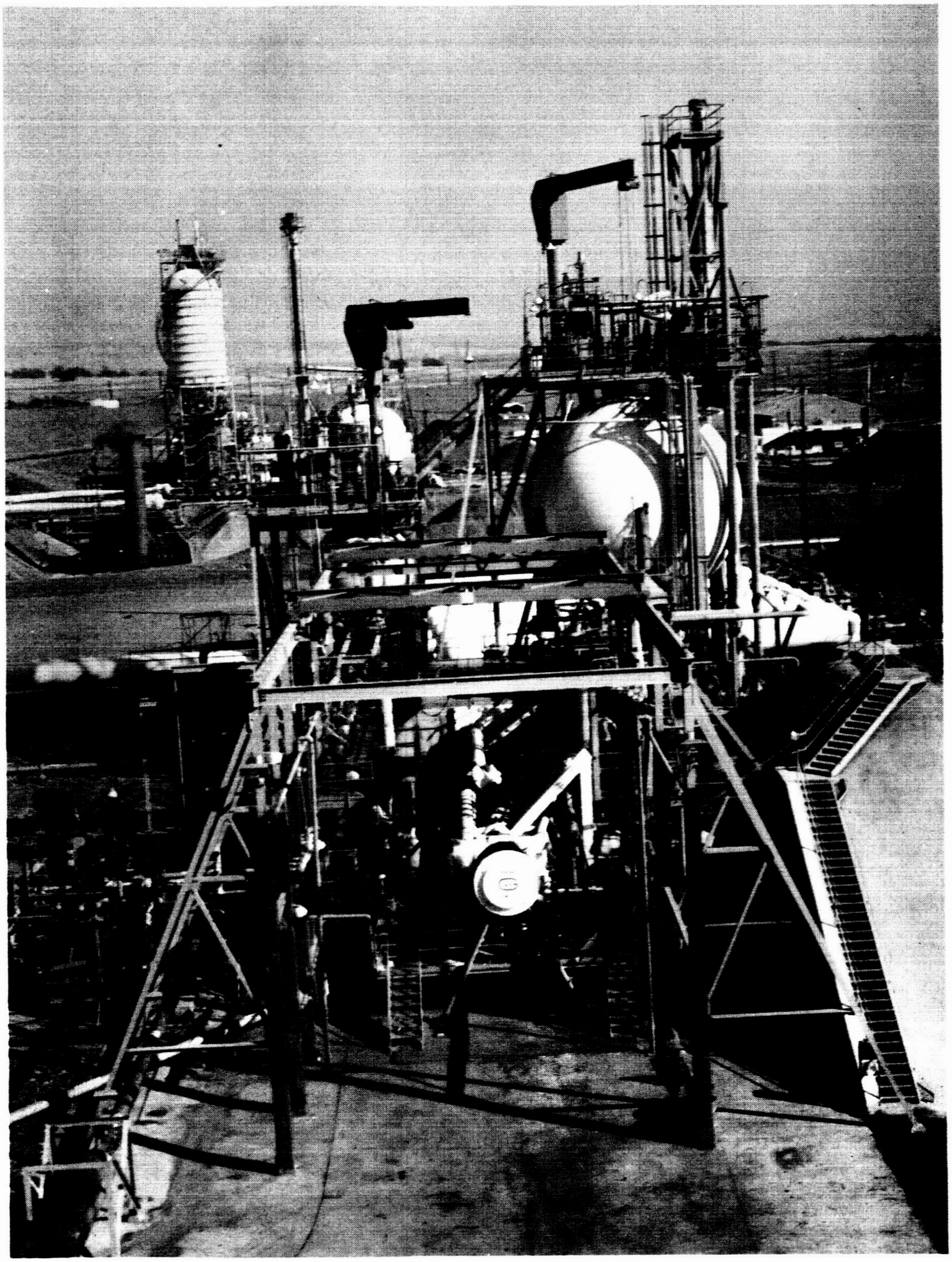


Figure 28

TEST STAND H-8 FACILITY

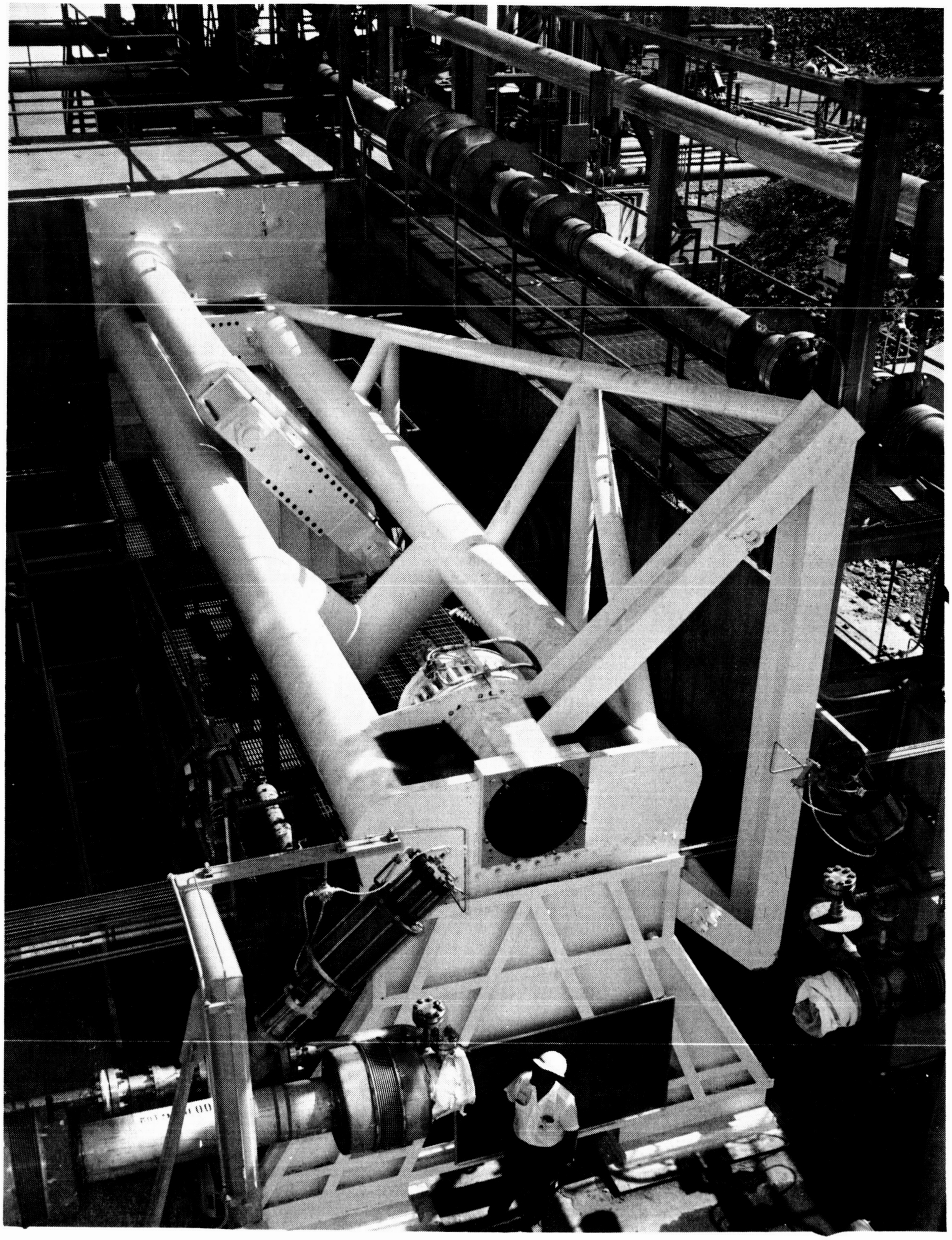


Figure 29
M-1 THRUST CHAMBER ASSEMBLY THRUST MEASURING SYSTEM

The ground for the thrust measurement system is provided by the two pipe columns, which extend from the thrust wall and are joined at their free ends with a cross-beam weldment. The free end of this ground system is stabilized by a vertical structure attached to the deck plate.

The only other major structural assemblies of the thrust system are the dummy gimbal support frame and the calibration system ground. The dummy gimbal support frame is actually an integral portion of the basic thrust frame and is comprised of a system of pipe columns and structural shapes welded to the south side of the thrust frame. This system is structurally designed and arranged to accept the gimbal actuator arms of the M-1 engine configuration. The calibration system ground is comprised of two tension rods extending from the thrust wall and tied at their terminal ends by a cross-beam weldment. The tension rods are each seven inches in diameter and are located inside the thrust system ground columns. This is done for convenience as well as to preserve structural compactness.

B. THRUST MEASUREMENT ASSEMBLY

The thrust measurement system, as originally conceived, consisted of a single BLH strain gage type load cell, Model U3XXA - 1,500,000, two flexure pivots, and necessary adapters. The load cell had been procured at the time that the thrust system design was initiated and the interface requirements were fixed. A flexure pivot was to be installed on either side of the force measuring load cell. These flexure pivots were to be structurally capable of reacting a load of 2,800,000 lb in tension at zero degrees angular deflection. The flexures were to be of the universal type whose force axes intersect at a common point in space regardless of angular deflection of the flexure. The rated capacity of the flexure as a function of the angular deflection was specified not to fall below the curve shown in Figure No. 30. Each flexure pivot was specified to have a shear load capacity of 10% (280,000 lb) of rated load. Wherever possible, all portions of the flexure were to operate below the endurance limit of the flexure material for rated load of the flexure. Also, local stresses in the highly stressed portion of the flexure were to operate below the 1,000,000 cycle fatigue strength of the flexure material. The flexures were required to interface with the pre-procured load cell and were to be provided with the appropriate adapters to mount the assembly in the thrust system. The adapters were designed to allow a quarter-inch lateral adjustment of the assembly centerline.

C. CALIBRATION ASSEMBLY

1. Description

The calibration system is comprised of a BLH load cell identical to the thrust cell, two flexure pivots and adapters, and a hydraulic piston with a 1,500,000 lb force capability. The connection between the hydraulic piston and the load cell/flexure assembly is through a clevis. The bore of the clevis-pin eye through the male portion of the clevis is elongated two inches. When the eight inch

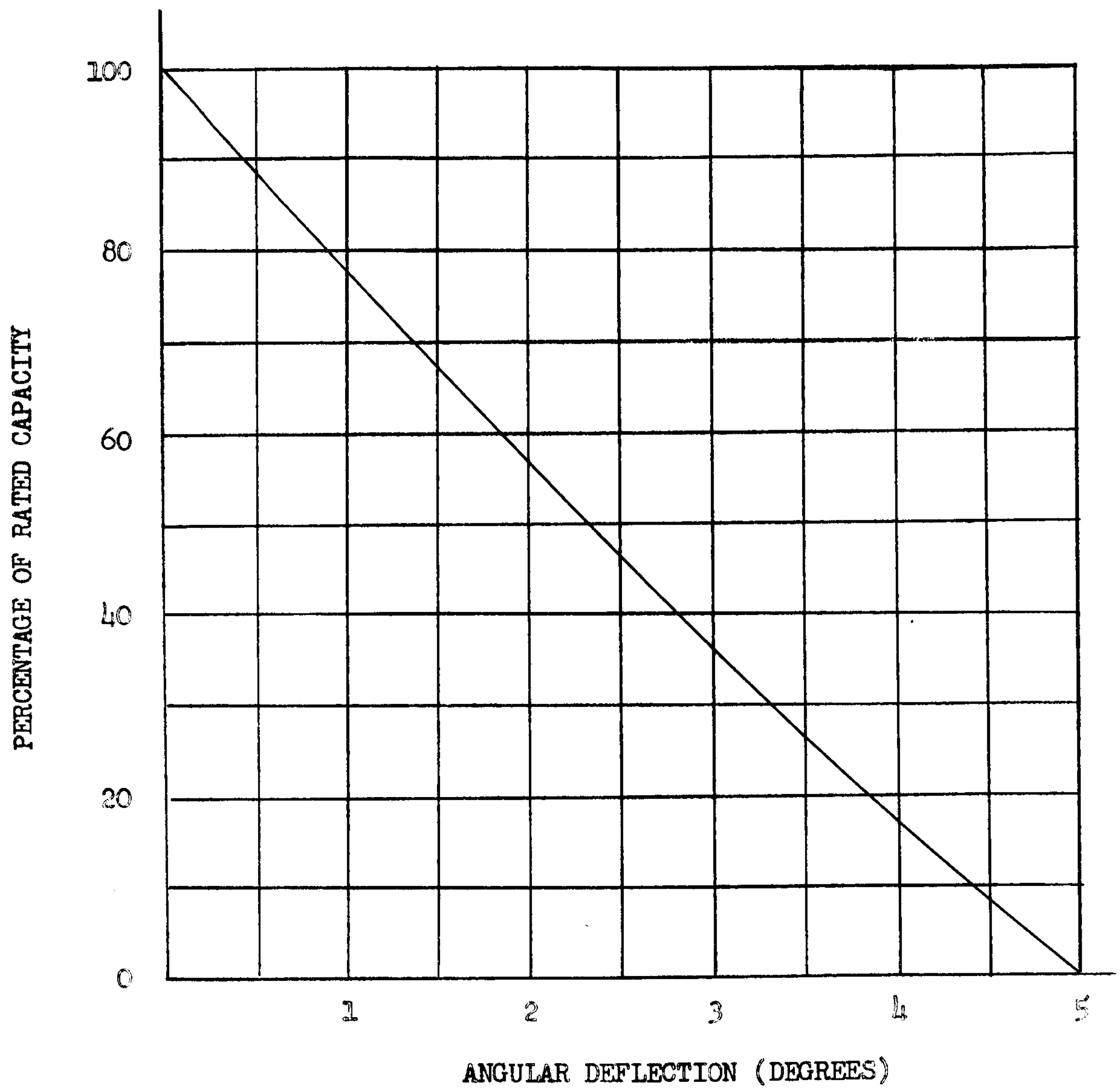


Figure 30

FLEXURE PIVOTS, RATED CAPACITY VS. DEFLECTION diameter clevis pin is centered in the elongated hole, no load can be transmitted through the calibration system. This provides a means for isolating the calibration system during a test firing without requiring the removal of any portion of the system, including the large and cumbersome clevis pin.

The hydraulic cylinder has a three inch stroke and is designed to accept 3500 psi on either side of the piston. The bore of the piston is 28.25-in. and the diameter of the piston rod is 10½-in. The net area of the rod end is 540-in^2, requiring a net pressure of 2800 psi to achieve a calibration load of 1,500,000 lb. The hydraulic cylinder is specifically designed to be mounted on the beam of the thrust measurement system. The cylinder tie-rods are screwed directly into this beam. The blind end of the hydraulic cylinder is not structurally capable of withstanding the internal pressure of the cylinder without being mounted to and supported by the face of the beam. The calibration system is controlled by a closed-loop servo system, which is described separately.

The independently-supported calibration system applies the calibration load to the back side of the thrust chamber assembly mounting beam and this load is transmitted through the thrust measuring system to the thrust wall in the same manner as the thrust chamber assembly thrust force. Independent support of the calibration system is a significant refinement of the standard design practice utilized for thrust systems of this size. Normal practice is to support the calibration system from some convenient member of the basic thrust measurement structure. Application of the calibration load in this manner results in some of the members of the thrust system not being loaded and some members of the system being loaded oppositely from what would occur during actual test conditions. However, independent support of the M-1 system ensures that little or no interaction occurs between the thrust measuring and calibration systems. Also, with one minor exception, the calibration load results in the same deflection and loading of the thrust system as would occur during static motor test firings. The single exception is that the massively reinforced concrete thrust block is not deflected during the calibration because the load is introduced in a "bootstrap" configuration. The deflection of the thrust block is not considered to be a significant factor.

2. Control

The control of the calibration system is provided by a five gallon per minute servo valve connected to the hydraulic cylinder. This servo valve is controlled by a closed-loop system. A signal from one bridge of the calibration load cell system is fed to the servo control amplifier located in the control room. A set point voltage is supplied to the servo amplifier through an external source. The internal voltage source is equipped with ten finely-calibrated potentiometers, which permit the accurate adjustment of ten voltage levels and a ten-position selector switch which allows for the selection of any of the ten voltages. The control voltage steps are normally adjusted to provide ten equal voltage increments which are equivalent to 1/10 of the full load voltage output of the calibration load. Any of the potentiometers can be adjusted to provide a no-load to full load control of the calibration system. The control system has been electrically designed so that the rate of correction (i.e. the rate of change of load of the calibration cylinder) is proportional to error signal (i.e. the difference between the set point voltage and instantaneous output voltage of the calibration load cell). This ensures that there is no overshoot or undershoot of the calibration load between load steps and that the actual calibration load becomes asymptotic to the desired load level in an exponential manner regardless of the load increment. This eliminates the effect of stand hysteresis from the individual calibration loads. For the same reason, a small orifice connects the high and low pressure side of the calibration piston, thereby requiring some finite flow of hydraulic fluid through the servo valve to maintain any desired calibration load. This assures a positive correction signal to the servo valve at all times and eliminates the hysteresis characteristic of the servo valve drive system.

The orifice for the H-8 thrust system calibration cylinder is a drilled hole through the piston; however, a better selection would have been an externally-mounted needle valve. The flow through the orifice at maximum load output of the calibration cylinder is equal to the flow capability of the servo valve and the full correction signal is required to maintain this load level. The current system can be readily modified by substituting a larger servo valve at such time as the combined flow through the orifice and leakage around the piston seals exceeds the flow capability of the existing servo valve. The needle valve would have eliminated this requirement or would have delayed the necessity for replacing the piston seals.

The current system is equipped with microswitches which indicate the no-load travel limits of the clevis pin within the elongated clevis eye. The microswitches operate panel-mounted lights in the control room. These lights indicate if the travel of the opposing segments of the clevis connection have exceeded the prescribed limits. In such instances, the travel of the calibration piston can be controlled by providing an external correction signal to the servo valve. This is necessary as there is no signal from the calibration load cell in the no-load condition and the calibration system drifts in the direction of the last correction signal. While this mode of operation has proven satisfactory for the static calibrations performed to date, it is a cumbersome procedure and does not provide a positive calibration system control. For this reason, the hydraulic lines to the servo valve are blocked by hand valves whenever it is desired to fix the position of the calibration system. The substitution of remotely-operated valves in the hydraulic lines will make the operation of the calibration system fully remote.

The ability to remotely control the entire thrust calibration system is required to perform immediate pre-fire and post-fire calibrations of the thrust system. Immediate pre-fire and post-fire calibrations are ideal in that the pre-fire calibration provides a valid calibration of the system at the time of the test and the post-fire calibration verifies that no change has occurred during the test. The normal Aerojet-General Corporation technique of periodic system calibrations is generally acceptable but the possibility of a system change caused by an unreported system modification between calibrations, or the restriction of some portion of the system as a result of some foreign object lodged in the thrust system could result in an invalid thrust firing.

The existing configuration of the M-1 propellant lines on Test Stand H-8, reduces the significance of the pre-fire and post-fire calibration capability. In particular, the deletion of the thrust chamber valves and the substitution of facility valves remotely located upstream of the thrust chamber assembly obviates the ability to perform a valid calibration of the test firing configuration because this would require that the propellant lines be chilled and pressurized. However, a pre-fire check to verify general system integrity is still planned.

D. FLEXURE PIVOTS

The four flexure pivots required for this system were procured as a subcontract to the basic contract for the fabrication of the thrust system. This particular contract arrangement was selected so that design changes could be made by the system fabricator as required to accommodate the interface requirements of the flexure pivots as well as to simplify the method of incorporating minor design changes found necessary during the shop drawing phase or fabrication. The thrust system contractor submitted a bid with several cost options based upon the final selection of a flexure pivot manufacturer. There were three companies that submitted bids for the four flexures, the Ormond Corporation, Santa Fe Springs, California; Alinco, Cumberland, Maryland; and Inca Engineering Corporation, San Gabriel, California. The first two manufacturers offered enlarged versions of their existing designs, both of which required the intricate machining of a large billet. The Inca design was a multi-piece assembly with a capability for replacing damaged or fatigued parts. While the Inca design concept was technically interesting and the cost was competitive, the design was not sufficiently proven at that time, even in smaller sizes, to warrant selection. Therefore, the Ormond Corporation was selected as the flexure manufacturer upon the basis of least cost as well as their extensive experience in the fabrication of flexure pivots for identical application, including the 3,000,000 lb rated flexures for use in the Saturn Program at the Marshall Space Flight Center. Figure No. 31 shows the configuration of the Ormond design.

Inasmuch as the flexure pivots represented nearly 40% of the total thrust system cost and became the major obstacle to the completion of the thrust system, the sequence of events leading to the deletion of the flexure pivots from the thrust system and the incorporation of two downgraded flexures in the calibration system is briefly summarized.

The Ormond Corporation manufactured four flexures in accordance with the specification. The first unit failed during a tension proof test at the Structural Testing Laboratory of the University of California. The test was to have consisted of a series of proof loads of 2,500,000 lb tension with the flexure deflected one-half degree. The flexure failed at a load of approximately 1,100,000 lb tension. Subsequent analysis of material samples removed from the flexure indicated that the material exhibited virtually no ductility near the center portion of the flexure body. This material was air-melt, Vasco-jet 1000 manufactured by Vanadium Steel Corporation of Latrobe, Pennsylvania. The original billet had been reduced 40% by forging, which

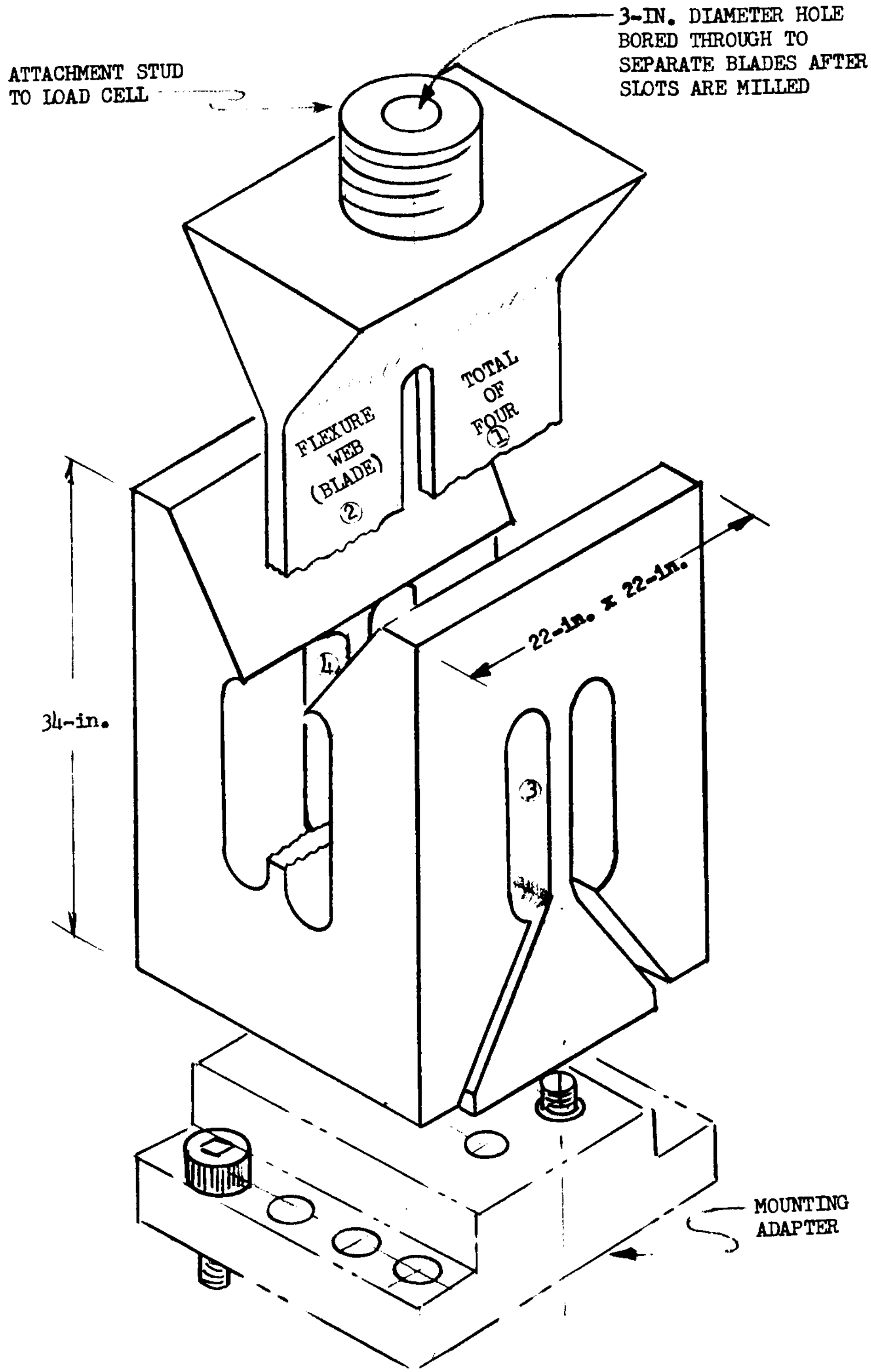


Figure 31

Flexure Pivot

was the maximum possible because of equipment limitations. The forged billet was then machined by Ormond and the machined flexures were heat-treated to an Rc 42-44 condition. Under normal circumstances, this material in this condition would be expected to exhibit 10% ductility (i.e. percent elongation).

A detailed inspection of the failed flexure revealed that the rupture of the flexure blades passed through an improperly machined radius fillet near the center of the flexure body. It was concluded that the lack of material ductility and the stress riser was the cause of the failure. It was also concluded that this lack of ductility resulted from insufficient forging action near the center of the large billet. Two of the remaining flexures were remachined to remove the stress riser and then reheat treated to Rc 34-37. Under these conditions, the material specimens from the center portion of the failed flexures exhibited a tensile yield strength of 167,000 psi and a minimum ductility of 3½%. These two flexures were no longer considered acceptable for use in the thrust measurement position because of the reduced tensile strength as well as the reduced endurance strength at the new heat treat. However, they were considered acceptable for the calibration system where they would not be subjected to shock loading, vibration or thrust transient overloads. The two flexures were successfully tested to a load of 2,125,000 lb tension at one-half degree deflections.

Ormond proposed to manufacture two new flexures for the thrust measurement position. These new flexures would have larger and less flexible load carrying blades and would be made of vacuum melt Vasco-jet 1000. The vacuum melting process reportedly improved the ductility characteristics of the steel and the increased load carrying cross-section reduced the operating stress of the highly stressed load carrying members. The heat treat was reduced to a minimum value consistent with the reduced operating stress. All of this was intended to improve ductility at the operating stress level. Forging equipment limitations made it impossible to improve ductility by increased forging.

The first of the two new flexures tested at the University of California failed at a load of 1,750,000 lb tension at an angular deflection of one-half degree. Inspection of the failed part indicated a brittle failure eminating from what appeared to be an insignificant stress riser at the center of the flexure body. It was concluded that the other flexure would suffer the same failure.

Testing schedules did not permit further manufacture of flexures. Also, the circumstances of the second flexure failure did not rule out the possibility that even a third design might fail because of a lack of knowledge regarding basic metallurgical properties of the center portions large steel forgings.

E. ROD FLEXURE

At this point, it was necessary to provide a replacement for the Ormond flexures for the thrust measurement position. The configuration of the completed thrust structure severely limited the length of any replacement flexures. The schedule did not allow for a rebid of the flexure portion of the thrust system. A single rod flexure was designed to substitute for the two flexure pivots. The criteria for the rod flexures were: a design load 1,500,000 lb tension; an L/D ratio of 15; and a safety factor of 2 based upon the yield strength at the design load.

The resulting flexure is shown in Figure No. 32. The rod was fabricated from 4340 steel and heat-treated to a hardness of Rc 40-44. New adapters were required to install this new thrust system configuration. While the rod flexure does not provide the bending moment and shear isolation of the flexure pivots, it does provide the necessary flexibility to accommodate the relative motion between the opposing mounting surfaces of the thrust system. The size of the H-8 thrust system and the nature of its support system minimized the lateral and rotational motion of the system. Therefore, the rod flexure was considered to be a reasonable compromise under these circumstances.

The rod flexure would be replaced with compound modular flexure pivots if schedules requirement and funding permit this improvement. The Inca Corporation has at least two alternative flexure designs which appear to be suitable and are now available. They built and tested a 200,000 lb scale model of their proposed multi-pieced flexure. The fabrication of the scale model resolved some incorrect basic design details that were apparent during the initial proposal evaluation. The test data indicate that mechanical "slop" and non-repeatable hysteresis characteristics are virtually non-existent. The very nature of the multi-piece design eliminates the Ormond flexure problem inasmuch as the blade and body portions of the flexure can be made of different materials selected to match the stress levels at that particular station. Alternatively, the Alinco Corporation now manufactures a flexure, designated the Mark X model, which consists of two concentric thick-walled cylinders joined at each end by welding. Each cylinder is machined from a separate forging and the flexure action is provided by slots milled through the walls of the cylinders. The major advantages of this design are that the highest stresses occur in material which has been effectively forged and that the less ductile material near the center of the billet has been removed. It is this latter consideration that could precipitate a failure of the entire part as observed with the Ormond flexures. As a result of experience, replacement flexures of the Ormond fabrication concept would be reviewed with prejudice.

IV. ANALYITICAL PREDICTIONS

After the thrust system structural design was completed, a second series of design analyses was started for the purpose of evaluating the probable performance of the thrust system under both normal and abnormal test conditions. To a large extent, the analyses were required to verify the performance of the system when subjected to: normal axial thrust, but with a rod flexure substituted for the Ormond flexure pivots; an oscillating non-axial load during the start transient; and a severe start transient axial thrust force. While not directly related to the analysis of the thrust system itself, an elaborate analyses was conducted to determine

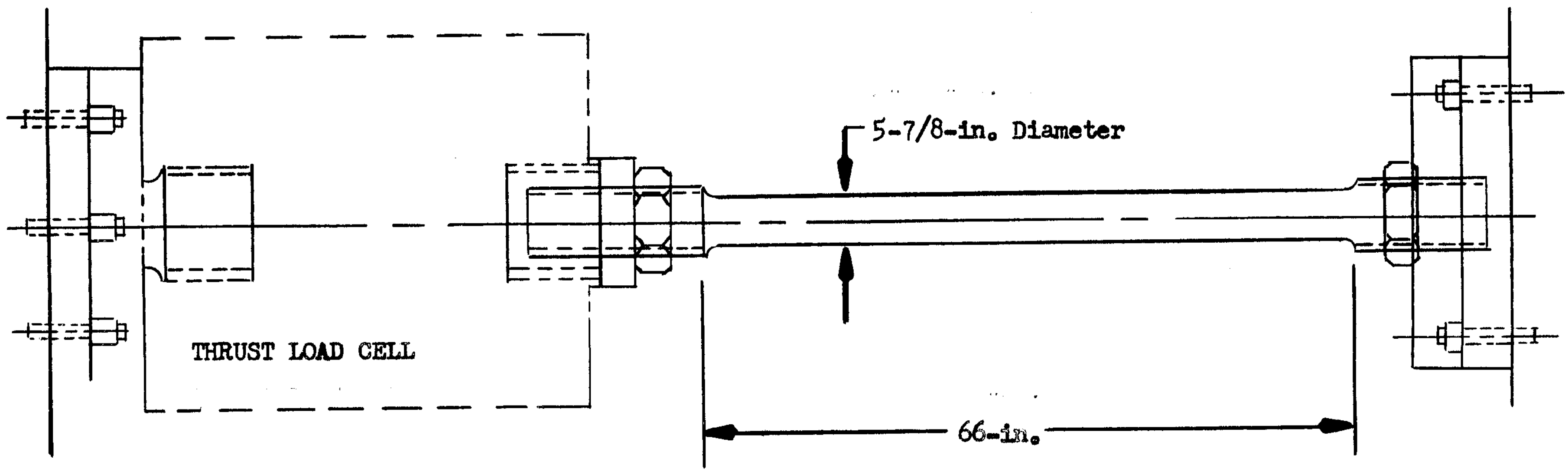


Figure 32

THRUST MEASUREMENT ROD FLEXURE

the dynamic reaction of the entire H-8 test facility. This analysis considered the dynamic characteristics of the thrust system and indirectly generated information of interest concerning the basic thrust system.

The analysis of the elastic deformation of the thrust system, whch subjected to 1,500,000 lb axial thrust, indicated the following system characteristics:

A.	Total axial deflection at 1,500,000 lb	.3-in.
B.	Angular rotation of the support flexures	
	1. Aft flexure (nearest thrust chamber assembly)	.33 degrees
	2. Forward flexure	.13 degrees
C.	Thrust measurement error caused by the weight component of the displaced pendulum	+ .025%
D.	Thrust measurement error caused by the restraint of plate flexures	- .07%
E.	Thrust measurement error caused by the angular misalignment of the thrust axis	Negligible

Each of the above static errors is a function of the system configuration and structural characteristics; therefore, they are repeatable and can be accurately compensated for by test stand calibration.

The non-axial start transient loading resulting from the asymmetrical expansion of the thrust chamber gases during the start transient required that an analysis be made of the non-axial vibration characteristics of the thrust system. The specific loading function is shown in Figure No. 26. The analysis and report of the thrust system reaction was published in an Aerojet-General AETRON Division study (1).

In summary, the referenced study indicated that only two modes of non-axial vibration were excited to any significant degree by the predicted forcing function. The first mode was the rigid body rotation of the thrust frame in a horizontal plane at 40 cps and the second mode of vibration was a 48 cps wracking (parallelogram distortion) of the thrust frame. Neither of these two modes was excited to the extent that structural modifications of the thrust frame were required. All other modes of non-axial vibration were at higher frequencies and were less affected by the forcing function.

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(1) Drew, J., Test Stand H-8, Start Transient Loading, AETRON Study H6017, May 1964

The axial dynamic characteristics of the thrust system were analyzed using the spring and mass computer model shown in Figure No. 33. The fundamental modes of axial vibration of the model were determined by matrix iteration. The thrust system was then analytically subjected to a "hard-start" (Figure No. 25), to determine the dynamic reaction and/or the thrust to transient loading. The steady state reaction of the thrust system to the random vibrational energy generated by the thrust chamber assembly combustion process was analyzed. The method of analysis was based on previous Aerojet-General Corporation analyses (2) and National Aeronautics and Space Administration studies (3).

The significant results of the dynamic analysis of the axial vibration of the thrust system are summarized in Table I.

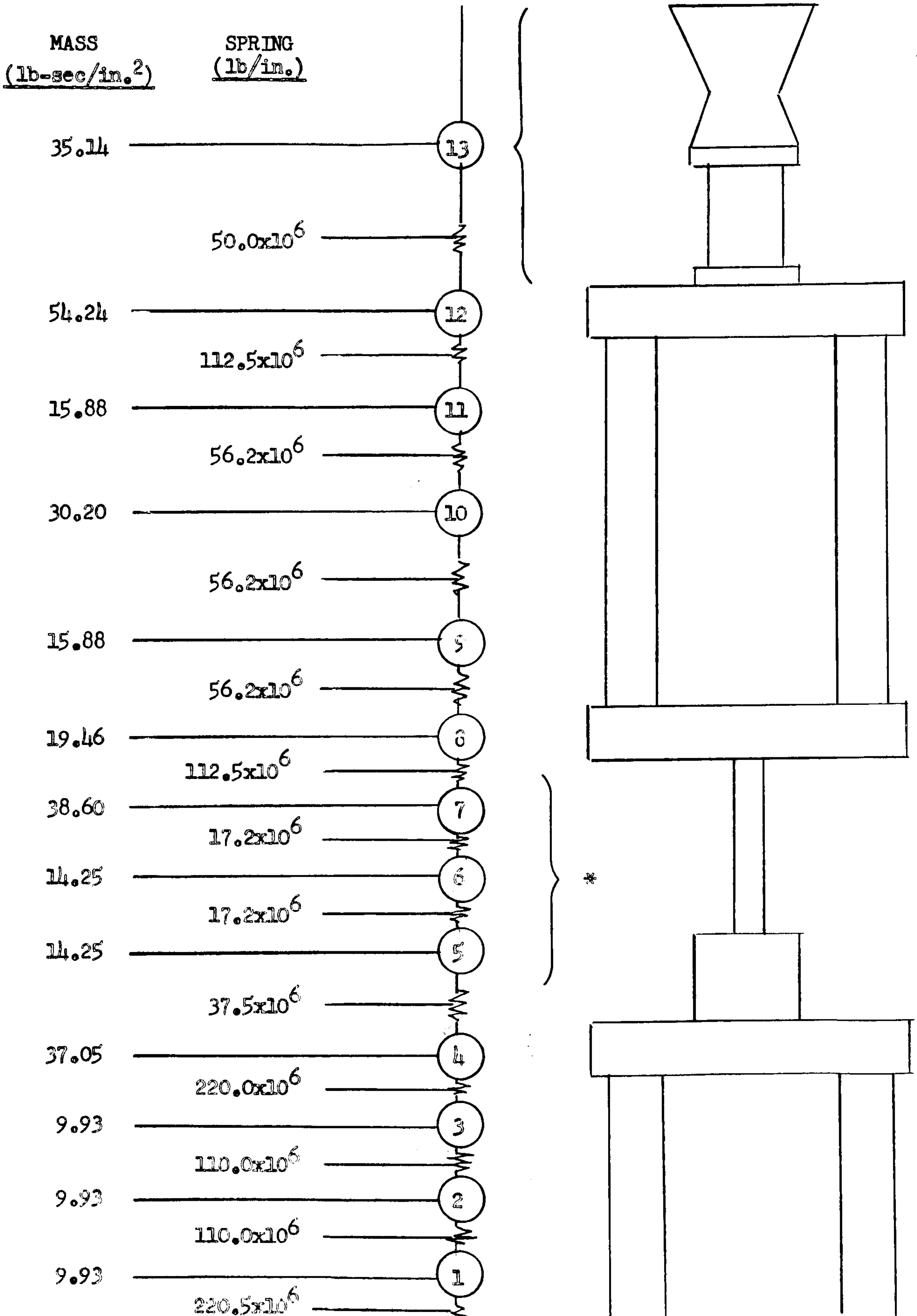
TABLE I

RESULTS OF DYNAMIC ANALYSIS

Free Vibration	
Natural Frequency	23.7 CPS - 1st Mode
	96.7 CPS - 2nd Mode
	136 CPS - 3rd Mode
Start Transient	
Hard Start	600K in 20 MS
Over Shoot	65% of 600K
Acceleration Loading of TCA	4.5 g's
Steady State Vibration	
TCA Vibratory Power	.2 g^2/CPS
Assumed System Damping Factor	3% of Critical
Acceleration Loading of TCA	4.8 g^2/CPS

(2) Cline, G. F. and Kessler, E. L., Estimated Dynamic Environments/Loads for M-1 Engine and Components, AGC Report No. 4-4-73R, 16 October 1964

(3) Barrett, P. E., Techniques for Predicting Localized Vibratory Environments of Rocket Vehicles, NASA TN 1836, MSFC, Huntsville, Alabama, October 1963



* Ormond flexure mass data used, rod flexure spring data

Figure 33

LONGITUDINAL DYNAMIC MODEL

V. CALIBRATION AND DATA REDUCTION

The calibration of the thrust measurement system consisted of two distinct phases; the static and the dynamic. These calibrations were entirely different in nature and to a certain extent the results were required for different purposes.

The static calibration of a thrust system is needed to determine system bias and system repeatability. The static calibration of the subject thrust system also provided necessary data for the calibration of the rod flexure strain gage system and provided deflection data of interest for future analytical studies. It also provided data for comparison with the predicted system characteristics based upon the analysis of the thrust system.

The configuration of the thrust system for the static calibration is shown in Figure No. 34. The basic test procedure consists of applying a load of 1,500,000 lb in a series of increasing incremental loads and then, returning to no-load in decreasing load increments. The previously described servo-controlled calibration system was used to load the thrust system.

It is necessary to duplicate as nearly as possible the test conditions to perform a valid static calibration of a thrust system. The ideal calibration configuration for the subject system would be with the thrust chamber assembly mounted, the propellant lines attached, the propellants bled-in, and the propellant lines pressurized to operating pressure. With the single exception of substituting liquid nitrogen in the fuel lines for reasons of safety, this configuration was used for the system calibration. Special blind flanges were inserted in the propellant systems at the facility/thrust chamber assembly interface so that the operating conditions of temperature and pressure could be maintained throughout all those portions of the propellant systems that are likely to affect thrust measurement.

To date, this particular configuration has been subjected to three series of increasing and decreasing incremental calibration loads. The analysis of these data is presented in Table II, but must be qualified because of the small number of tests performed. However, it appears that the results are indicative of the stand capability and that these preliminary results will be verified by future calibrations. The indicated calibration results are a statistical analysis of the increasing load data in the range of 1,500,000 lb to 1,350,000 lb force and take into account the calibration data of the two load cells, the load cell output of the two cells during calibration, and the data acquisition system error. The percent bias (-0.04%) represents the numerical factor by which the steady-state thrust data obtained from a test firing of the M-1 thrust chamber assembly will be corrected. The percent error figure indicates that 99.7% of the reduced data will be within $\pm$ 0.18% of the true value.

In addition to the described system calibration, a series of static calibrations were initially planned wherein the system configuration would be modified between each calibration sequence. The purpose of these additional calibrations would be to determine the source, nature, and magnitude of the thrust system restraints by systematically isolating the various contributions to system restraint and non-repeatability. This data would be of use in identifying the need for system modifications as well as providing valuable information for the design of future systems.

TABLE II

SUMMARY H-8 CALIBRATION DATA

I.	Overall Thrust System Accuracy (Average of Two Bridges)		
	A. Bias		-.04%
	B. Non-Repeatability		$\pm$.06% (1σ)
II.	Load Cell Correction (S/N 32460)		
	A. "A" Bridge	+.05%	$\pm$.07% (1σ)
	B. "B" Bridge	+.03%	$\pm$.09% (1σ)
III.	Total Random Error Data		
	A. Revere Dead Weights		$\pm$.006% (3σ)
	B. Multiply Arm Ratio		$\pm$.100% (3σ)
	C. Lab. Cal. S/N 32459 (Shunt to Force)		
	1. "A" Bridge		$\pm$.040% (1σ)
	2. "B" Bridge		$\pm$.028% (1σ)
	3. Average		$\pm$.024% (1σ)
	D. Thrust Bias (S/N 32460)		
	1. "A" Bridge	-.047%	$\pm$.078% (1σ)
	2. "B" Bridge	-.030%	$\pm$.097% (1σ)
	3. Average	-.0385%	$\pm$.062% (1σ)
	E. System Ambiguity		$\pm$.025% (1σ)
	F. Dynamic Error (Estimate)		$\pm$.050% (1σ)
	G. Dual Bridge Average		$\pm$.093% (1σ)
	H. Single Bridge Measurement		$\pm$.127% (1σ)

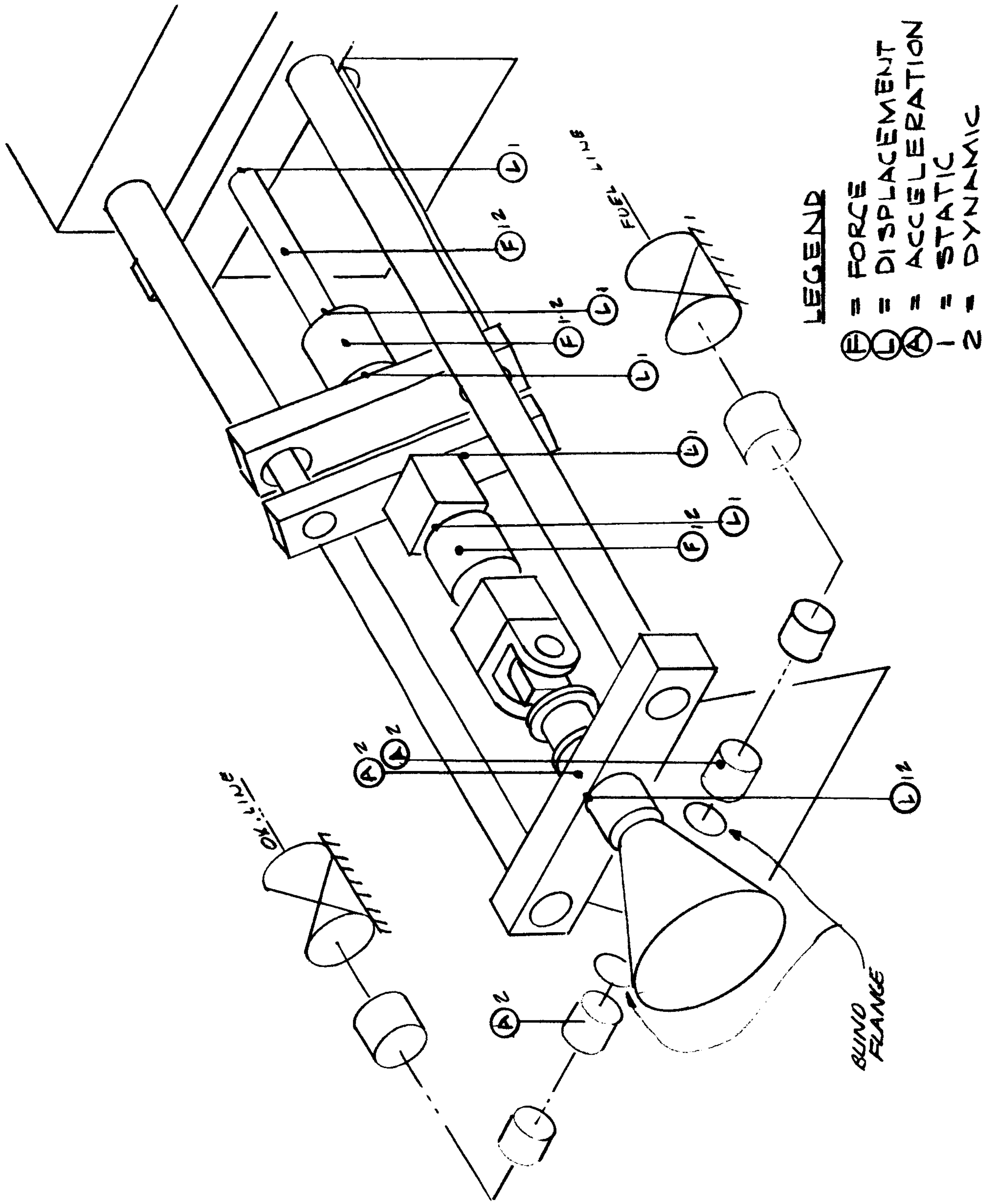


Figure 34

Calibration Configuration

The basic system configurations planned are the thrust chamber assembly mounted without the propellant lines attached; the thrust chamber assembly mounted with the propellant lines attached, but at ambient temperature and pressure; the thrust chamber assembly mounted with the propellant lines attached and pressurized at ambient temperature; and the thrust chamber assembly mounted with the propellant lines attached and chilled but not pressurized. Actually the last configuration was tested prior to the system calibration previously described inasmuch as the preparation was identical. However, the data is not cited in this report because it is not significant by itself. If repeated system calibrations continue to repeat the excellent system performance indicated by the first series of calibrations, the above will not be calibrated separately.

As stated, the static calibration results will be used to reduce steady-state thrust data. The system bias factor will be applied to raw thrust load cell data that has been electrically filtered. The first series of thrust chamber assembly tests performed with the system will be exploratory in nature and of short duration. During these short duration tests, there will be no period of steady-state combustion and it is partially for this reason that the thrust system was dynamically calibrated (see Figure No. 34).

Dynamic calibration of the thrust system characterizes the thrust system so that transient force data can be analyzed. Dynamic calibration of the subject thrust system was also utilized to confirm the results of a structural dynamic analysis of the thrust system, thrust chamber assembly, and associated propellant piping. Reduction of transient thrust data requires that an analysis be made of the raw thrust data to separate the thrust input of the thrust chamber assembly from the dynamic response of the spring, mass, and damper system which comprises the basic thrust system. To provide the data for such an analysis, the thrust system was excited with a load impulse and load cell reaction to this forcing function was recorded. The forcing function was introduced by applying a pre-load to the system and instantaneously removing the load. In this manner, the force measurement system natural frequency and damping ratio were established. Also, the system linearity was observed.

The calibrating system provided an ideal means for applying a known input load to the thrust system. It was only necessary to replace the clevis pin in the force calibration string with a link designed to rupture at a prescribed load level. It was determined that a system pre-load of 200,000 lb to 300,000 lb was desirable. The normal clevis shear pin was replaced with a tool steel pin of smaller diameter supported within the clevis in such a manner that it would fail in bending (see Figure No. 35). The pin was heat-treated to Rc 50 and a small stress riser was machined at mid-span to induce a brittle failure at this point. An unsuccessful attempt to fail the pin during the first dynamic calibration test indicated that the pin was too ductile and all subsequent tests were satisfactorily performed by pre-chilling the frangible pin to liquid nitrogen temperatures.

Four dynamic calibration tests were performed to define the thrust system characteristics for the purpose of transient thrust analysis and to determine the

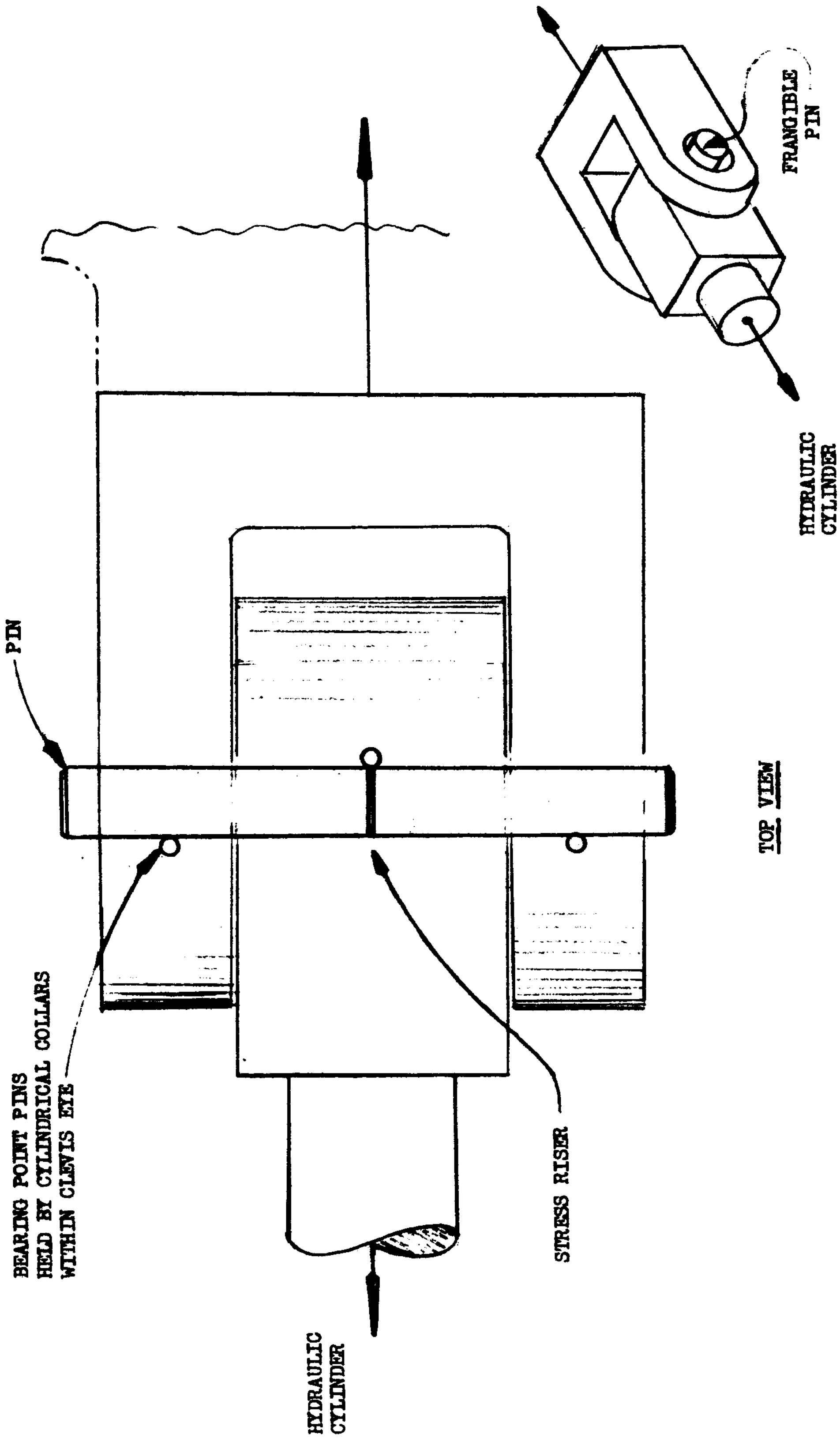


Figure 35

Dynamic Calibration Link

effectiveness of a system of propellant line dampers. These dampers were designed to eliminate excessive interaction loading of the thrust chamber assembly by the propellant piping as a result of relative motion between the thrust chamber assembly mounted on the thrust system and the propellant piping. The test results repeated satisfactorily and characterized the system.

Figure No. 36 is a reasonable copy of the typical load cell signature from the dynamic calibrations and shows that the system dynamics are characterized as a single degree of freedom spring mass system with a small damping factor. Thus, the thrust measurement system can be mathematically described by the simple equation:

$$F(t) = M(\ddot{x}) + B\dot{x} + Kx$$

Where: $F(t)$ = Thrust

M = Effective mass of system

B = Damping Coefficient

Kx = Load Cell Reading

However, expressed in this form, only the Kx term derived by the load cell is directly measured during a test. Mass, acceleration, and velocity terms would have to be generated by implicit means. The mechanics of the mathematical solution of the above equation is much more readily performed using Z-transform notation and finite difference numerical compensation techniques (4). To implement these techniques, the test force analog data is recorded on wide band magnetic tape for subsequent digitizing. A digital band limiting technique is used to smooth the raw recorded data points and prevent frequency folding into the primary data band. Thus, the filtered data are essentially undistorted in the primary frequency band of interest, but contain no components above the pre-determined cutoff frequency. The digital computer is then programmed with a numerical recursion formula that has, as a transfer function, the exact inverse of the dynamic measurement system transfer function. Thus, the product of measurement system dynamics and digital computer recursion formula yields a transfer function of unity over the primary band, as defined by the digital filter cutoff frequency. Because the filtered digital input data contain no high frequency components, the high frequency noise amplification properties of digital differentiation are not significant. The thrust system bias factor is then applied to the reconstructed thrust data.

Transient thrust data obtained by the aforementioned technique is limited to approximately ± 1%. This lower accuracy is a result of the number of data handling techniques and the inexactitude of the mathematical model.

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(4) Langill, Jr., A. W., Digital Reconstruction of Solid Rocket Motor Measurement Transients, ISA Proceedings, April 7 through 11, 1964, pp 57-65

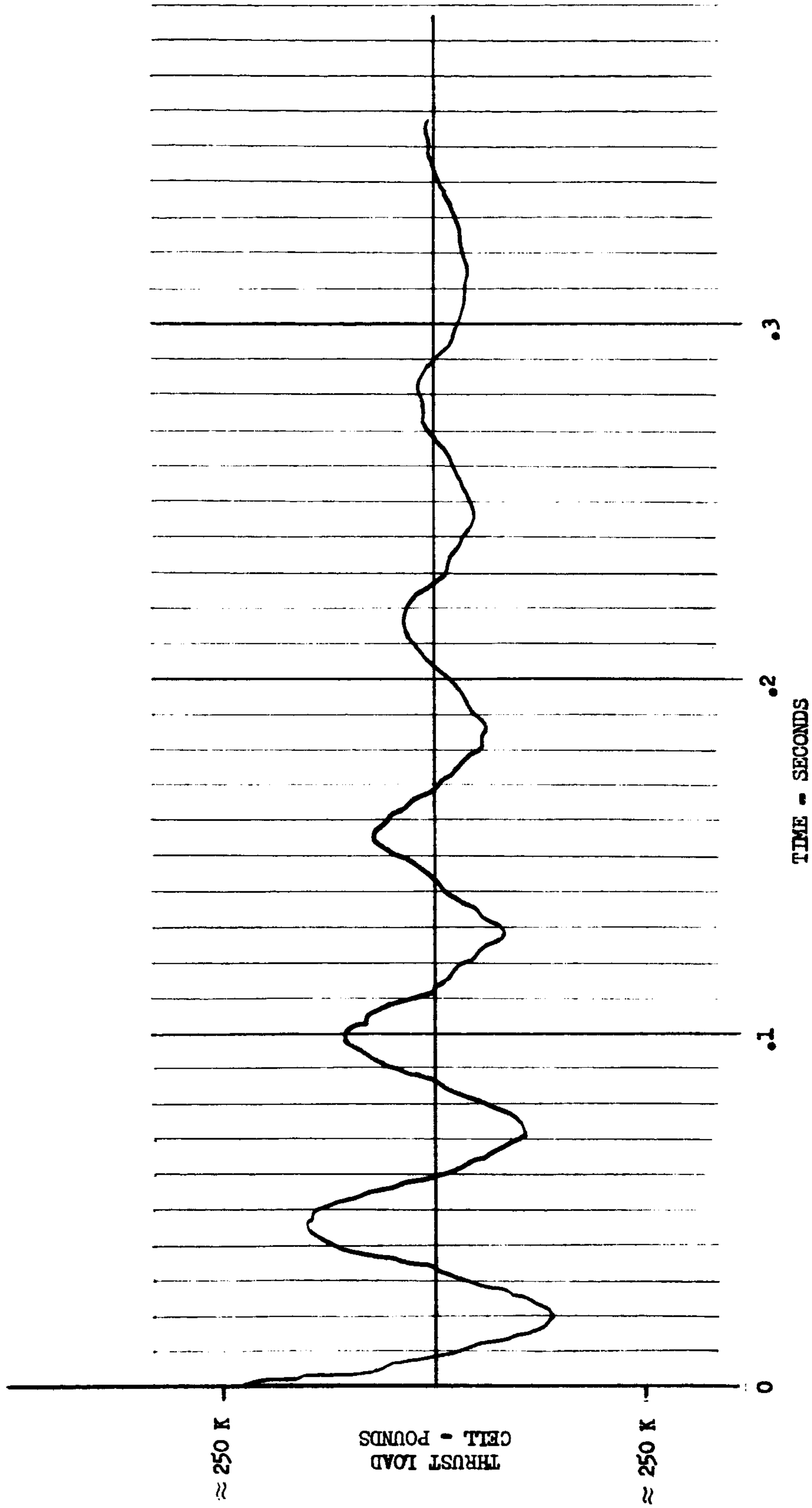

Figure 36

M-1 Thrust System Free Vibration Characteristics

VI. CONCLUSIONS

The M-1 thrust chamber assembly thrust measurement system, as built, meets or exceeds all design criteria. The results of the limited static calibrations that the thrust system will yield useful transient thrust data.

Rigorous dynamic analyses indicate that the thrust measurement system can react all predicted dynamic loading without damage and that the thrust system does not impose excessive loads upon the thrust chamber assembly or its associated systems. These analytical conclusions have been confirmed by the dynamic calibration data.

The M-1 thrust measurement system has demonstrated the feasibility and practicality of remote calibration for a system of this size and thrust by a closed loop servo-control system. Finally, the fabrication of the pivot flexures for the M-1 thrust system has identified a serious limitation of the single-piece flexure fabrication concept.

VII. RECOMMENDATIONS

The above conclusions apply to the system as designed and built in 1963. The following deviations to the H-8 thrust measurement system would be suggested for consideration in the design of a new system for similar application:

A. Utilization of a compression thrust measurement system of equivalent measurement accuracy. A compression measurement system significantly simplifies the details of structural design. However, it is necessary that special attention be given to minimizing lateral motion of the system and that a "shear" cell recently developed by BLH be used or that flexure pivots of maximum flexibility be used on either side of the measurement cell if a compression system is to have the accuracy and repeatable characteristics of a tension system.

B. Minimize the thrust load path through the thrust system structure to ground. The shortest possible load path reduces deflections and minimizes the moving mass of the system, thereby increasing stand frequency. The incorporation of a compression measurement system is generally compatible with this design objective.

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APPENDICES

APPENDIX A

SPECIFICATION FOR FORCE MULTIPLYING SYSTEM

(Extension of the Capacity of Existing Load Cell Calibration System)

May 1962

SPECIFICATIONS FOR FORCE MULTIPLYING SYSTEM
(Extension of the capacity of existing
load cell calibration system)

1. SCOPE

1.1 It is the intent of the force multiplying system specified herein to apply compression and tension calibrating forces to force measurement devices in specified increments up to 1,500,000 pounds. The force multiplying system shall extend the range of an existing 60,000 pound dead weight machine located in Building 3308, Liquid Rocket Plant, Aerojet-General Corporation, Sacramento, California.

1.2 The existing machine is capable of loading in compression and tension. It is a free hanging dead weight calibrator capable of loading in 20% increments over the following ranges: a) 5,000 pounds; b) 10,000 pounds; c) 20,000 pounds; and d) 50,000 pounds; One extra 10,000 pound weight can be lifted to provide a total capacity of 60,000 pounds. The above weights have been calibrated by the National Bureau of Standards to ± 0.005% of value.

1.3 The operation of the 60,000 pound dead weight machine shall remain the same with unimpaired accuracy. Moreover, the time required to change from the 1,500,000 pound system to the 60,000 pound system shall be kept to a maximum of one hour.

1.4 The force multiplying system shall use the existing control module of the 60,000 pound dead weight machine. Therefore, any additional control system components shall be incorporated into the existing control console and shall be operated from its present location.

1.5 FACILITY

1.5.1 The facility to house the force multiplying system will be furnished by Aerojet-General Corporation. The requirements of the housing facility shall be specified in the quotation submitted in response to these specifications. These shall include requirements for dimensions and modifications of existing structures. In particular, the Seller shall specify dimensions, materials and exact configurations of all support and force reaction pads.

1.5.2 The housing facility will be maintained by Aerojet-General Corporation between 65° and 85° Fahrenheit.

1.6 INSTALLATION

1.6.1 The installation of the force multiplying system shall be the Seller's responsibility. Any modifications including re-orientation of the present installation of the existing dead weight machine required for the incorporation of the force multiplying system shall be considered part of this responsibility.

1.6.2 The Seller shall provide adequate field engineering supervision during installation.

2. RESPONSIBILITY FOR SYSTEM DESIGN

2.1 The Seller shall be responsible for the effectiveness of the force multiplying system design. The responsibility to be assumed by the Seller shall include the compatability of the force multiplying system with the existing equipment and housing facility requirements specified by the Seller under paragraph 1.5.1.

3. REQUIREMENTS

3.1 The force multiplying system shall meet all the requirements of this specification.

3.1.1 Deviations from these specifications may be made only after written authorization is obtained from the Measurement Engineering Department, 8770, Liquid Rocket Plant, Aerojet-General Corporation, Sacramento. No increase or decrease in price is authorized except when executed as a formal purchase order change.

3.1.2 CODE CONFORMANCE - All electrical components, material, and details of construction shall conform with the requirements of the latest issue and revisions of the National Electrical Code of the National Board of Fire Underwriters, and the National Electrical Safety Code of the United States Department of Commerce.

3.1.3 DESIGN AND WORKMANSHIP - The design and construction of all components, the assembly of the components, and associated control systems shall reflect the most modern and accepted practices for this type and class of apparatus. The equipment shall be new. The work shall be executed in the best and most workmanlike manner by qualified technicians and mechanics in strict accordance with these specifications.

3.1.4 DESIGN DRAWINGS - Four reproducible copies of all system components, schematic drawings, outline dimensional drawings, control circuit diagrams showing all terminal connections, specifications, test and calibration procedures prepared by Seller shall be submitted to the Buyer for approval within sixty days after award of contract. One print of each drawing will be returned to the Seller with the Buyer's approval, or suggested changes noted. The Buyer's approval will be for the general design only and will not relieve the Seller of the responsibility of providing a complete system meeting the requirements of these specifications and fitting neatly, securely and accurately into or onto the specified space or existing dead weight machine.

3.1.4.1 The final design drawings, calibration and test procedures will be approved by Measurement Engineering Department, 8770, Liquid Rocket Plant, Aerojet-General Corporation, Sacramento, California, prior to commencing fabrication.

3.1.5 INSTRUCTION MANUALS - Ten sets of instruction manuals shall be provided with the equipment. Manuals shall contain all information required for technicians to operate, maintain, and calibrate the equipment properly. Physical layout and functional drawings shall be prepared in sufficient detail to provide technical understanding of the system. Methods of diagnosing troubles and other service information developed by the Seller shall be included. The manual shall include a suggested spare parts list.

3.1.6 WARRANTY - The Seller shall guarantee the performance of the equipment to meet or exceed the requirements of this specification and, in addition, shall guarantee all equipment against defective materials and workmanship for a period of one year from the acceptance date of the work. In the event that defects become apparent within this period, the Seller shall without delay repair or replace in the field such defective units or parts thereof as may be required at no additional expense to the Buyer. During this warranty period, in the event a malfunction in any component in the force multiplying system supplied by the Seller results in the system remaining inoperative for a period of more than five days, the Buyer reserves the right to initiate corrective action including services of contractor personnel. The costs of such corrective action shall be chargeable to the Seller under the terms of this warranty.

3.1.7 QUOTATION FORMAT - The Seller's quotation for the system referred to in these specifications shall comprise five items, each quoted separately: Item (1) shall deal with the basic machine; item (2) shall deal with the control system; item (3) shall deal with the alarm system; item (4) shall deal with installation, checkout and calibration at the Buyer's facility; item (5) shall deal with the recalibration at a six month interval as per paragraph 4.4.

3.1.7.1 The Seller's quotation will be evaluated by the Measurement Engineering Department, 8770, Liquid Rocket Plant, Aerojet-General Corporation, Sacramento, California.

3.1.8 SERVICE - The Seller shall state his capability to furnish field service after the expiration of the warranty. The Seller shall describe his service capability with regard to field office, technical personnel, calibration facilities and per diem or contract rates. Acceptance of these services shall remain the option of Aerojet-General Corporation and consequently are not to be included in Seller's quotation for the force multiplying system.

3.2 TECHNICAL DETAILS

3.2.1 CAPACITY - The force multiplying system shall produce a maximum force of 1,500,000 pounds.

3.2.2 LOADING - The force multiplying system shall be capable of loading a force measuring device in compression and in tension.

3.2.3 LOADING INCREMENTS - The force multiplying system shall multiply, at a rate of 25:1, all the load points produced by the 60,000 pound dead weight machine. (Refer to paragraph 1.2)

3.2.4 PIVOTS - Any pivots used in the force multiplying system shall be guaranteed by the vendor to perform satisfactorily as specified herein during a working life of 12,000 full load cycles, or one year, whichever occurs first. During their working life, the pivots shall not degrade the system to such a degree that the combined error of the system exceeds the value specified in paragraph 3.2.9.

3.2.5 MULTIPLICATION RATIO - The force multiplication ratio shall be 25:1 ± 0.1% over any selected range within the temperatures specified in paragraph 1.5.2. The force multiplying machine shall be adjustable to maintain the multiplication ratio within these limits.

3.2.5.1 The multiplication ratio is defined as the ratio of the effective lengths of the lever arms.

3.2.6 SENSITIVITY - The sensitivity of the force multiplying system shall be at least ± 0.01% of full scale of the selected range or better.

3.2.6.1 Sensitivity is defined as the minimum load, applied at the weight loading point, necessary to cause the multiplying arm to become measurably unbalanced.

3.2.7 REPEATABILITY - The repeatability of the force multiplying system on consecutive applications of full load shall be ± 0.02% of full scale of the selected range or better.

3.2.7.1 Repeatability is defined as the ability of the force multiplying system to reproduce a given force at the force application point on consecutive loading when approached in the same direction under the same conditions.

3.2.8 HYSTERESIS - The maximum difference between any two points on the hysteresis curve determined by the same dead weight loading of the multiplying machine shall be no greater than ± .02% of full scale of the selected range.

3.2.8.1 The hysteresis curve is defined as the locus of points on a weight vs. applied load to the force sensor diagram generated by sequential loading from zero to full scale of the selected range and subsequent return to zero loading in the reverse sequence.

3.2.9 SYSTEM ACCURACY - The combined effect of weight error, multiplication ratio, sensitivity, repeatability and other extraneous errors shall not exceed ± 0.1% of full scale of the selected range.

3.2.10 CONTROL - The force multiplying system controls shall include: a) A load selector switch to provide the same increments as the existing system except multiplied by the 25:1 ratio; (See paragraph 1.2) b) Provision for sequential loading in 20% increments over any selected range; c) Visual display of the load applied to the force sensing device during the entire loading sequences.

3.2.11 LOAD ALARM - An alarm system shall be incorporated in the multiplying system to signal any extraneous occurrence which could cause an error in the actual load applied to the force measuring device. This alarm shall signal: a) Weights not hanging free and clear; b) Force multiplying arm not in a level position; c) Any counter balance weights not hanging free and clear.

3.2.12 LOADING FRAME - The loading frame shall be adjustable vertically from 4 feet minimum to 9 feet maximum by means of electrically driven motors. The minimem width shall be 4 feet. These dimensions are subject to change and shall be confirmed by Aerojet-General Corporation with the Manufacturer of load sensing devices of the capacity that are to be calibrated in the force multiplying system.

3.2.13 SAFETY - Load applying members shall have a minimum safety factor of two. Stationary frames shall have a minimum safety factor of four. The force multiplying system shall be designed so that, in the event of a pivot failure duriing calibration of a force transducer, the supports and frames shall preclude any damage to the force transducer, loading frame and supports.

3.2.14 POWER CONNECTIONS - The force multiplying system shall include any equipment to generate and distribute hydraulic power as required in excess of that available in the existing calibration system. The method used to connect hydraulic and electrical lines to the loading components shall not introduce extraneous errors into the calibration system.

3.2.15 MECHANICAL STABILITY - The force multiplying system shall produce oscillation - free readout of any applied load within 120 seconds after application of the weights.

4. ACCEPTANCE REQUIREMENTS

4.1 FINAL ACCEPTANCE - Final acceptance shall be made during the operational checkout performed by Seller's Representative in the presence of the Buyer's Representative at the Buyer's facility. In the event of rejection due to non-compliance with the specifications, Seller shall repair or replace the rejected item to the satisfaction of and at no additional expense to the Buyer.

4.2 CALIBRATION - The calibration shall be performed at the Buyer's facility and will include the following: a) Determination of multiplication ratio at all possible loading points; b) Demonstration of the repeatability and sensitivity under loaded condition at all possible loading points. The demonstration acceptance tests shall comprise a minimum of ten cycles of operation, each cycle to include all loading increments as specified in paragraph 1.2.

4.3 CONFORMANCE LETTER - The Seller shall issue a letter of conformance upon completion of acceptance calibration.

4.4 RECALIBRATION - The Seller shall repeat the calibration tests as described in paragraph 4.2 six months after the initial acceptance of the system. This shall be considered as part of the Warranty as per paragraph 3.1.6.

APPENDIX B

CALIBRATION OF WEIGHTS

C O P Y

UNITED STATES DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS
WASHINGTON 25, D.C.

Test No. 2.6/165091
P. O. No. 5379

NATIONAL BUREAU OF STANDARDS
TEST REPORT
on
Fourteen Special Weights
and One Frame Assembly

Maker: Fairbanks, Morse & Co
Submitted by: Revere Corporation of America
Wallingford, Connecticut
For: Aerojet-General Corporation
Nimbus, California

Fourteen large weights of special design, and one frame assembly, were compared with standards of the National Bureau of Standards at the Clearing, Illinois, Master Scale Depot. As released, each weight was accurate well within ± 0.005% of its nominal weight. Although the weights do not conform to NBS Class C specification, the closures to the adjusting cavities of each weight do conform in general to Class C requirements. It was noted, however, that several cover plates within which the closure to the cavity was located were not securely fastened to the bodies of the weight.

The group consisted of the following:

Quantity	Item	Identification No.	Nominal Value
5	Weight	1-5	1,000 lb
3	"	6-8	2,000 "
2	"	9-10	4,000 "
4	"	11-14	10,000 "
1	Frame Assembly	-	1,000 "

The National Bureau of Standards uses the following relationship between the metric units of mass and the U. S. customary units of mass: One pound (avoirdupois) equals 0.45352937 kilogram.

For the Director,

s/H. H. Russell

H. H. Russell
Chief, Scale Unit
Mass and Scale Section
Metrology Division

C O P Y

U.S. DEPARTMENT OF COMMERCE

NATIONAL BUREAU OF STANDARDS

Address Reply To
NATIONAL BUREAU OF STANDARDS
WASHINGTON 25, D.C.   August 17, 1961

In Your Reply
REFER TO FILE NO.
2.6

Mr. John J. Robinson
Chief Project Engineer - Research
Revere Corporation of America
Wallingford, Connecticut

Dear Mr. Robinson:

Since our telephone conversation we have received from the Bureau representative at the National Bureau of Standards Master Scale Depot a very comprehensive report on the slight defect referred to in our report on the test of a group of large weights submitted by Revere Corporation of America.

The report as written is factual; in some cases the cover plate for the adjustment cavity was not securely welded or brazed to the body of the weight. This fact was not apparent, however, being indicated only during the process of flatting the seals by breakage of the paint at the line of juncture between the cover plate and the body of the weight.

These weights were protected from the elements during shipment to California and will be housed at all times during use. Moreover, the circumstances of usage will not subject the cover plates to impacts that might loosen them additionally. Consequently, our representative decided that slight mechanical fault would not impair the constancy of the weights beyond the anticipated inconstancy of weights of those denominations.

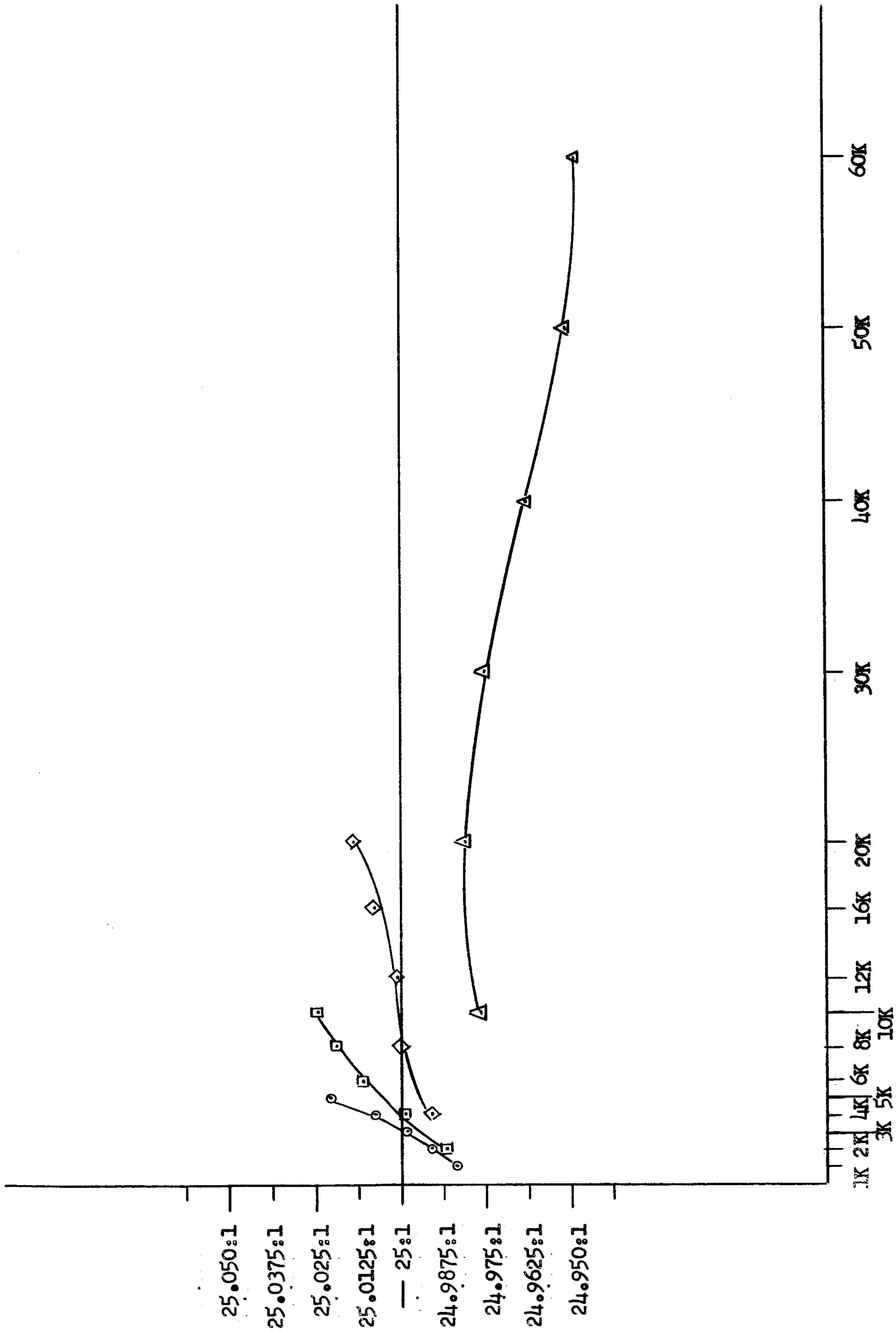
The writer agrees with this opinion but nevertheless recommends as a precautionary measure than an additional coat of clear lacquer or varnish be carefully applied over the line of juncture between the cover plates and body castings. It is inconceivable that this coat of lacquer could alter the weight value by an amount approaching the stated precision of weighing.

Sincerely yours,

S/H. H. Russell
Chief, Scale Unit
Mass and Scale Section
Metrology Division

APPENDIX C

CALIBRATION RESULTS OF CELL METHOD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25:1
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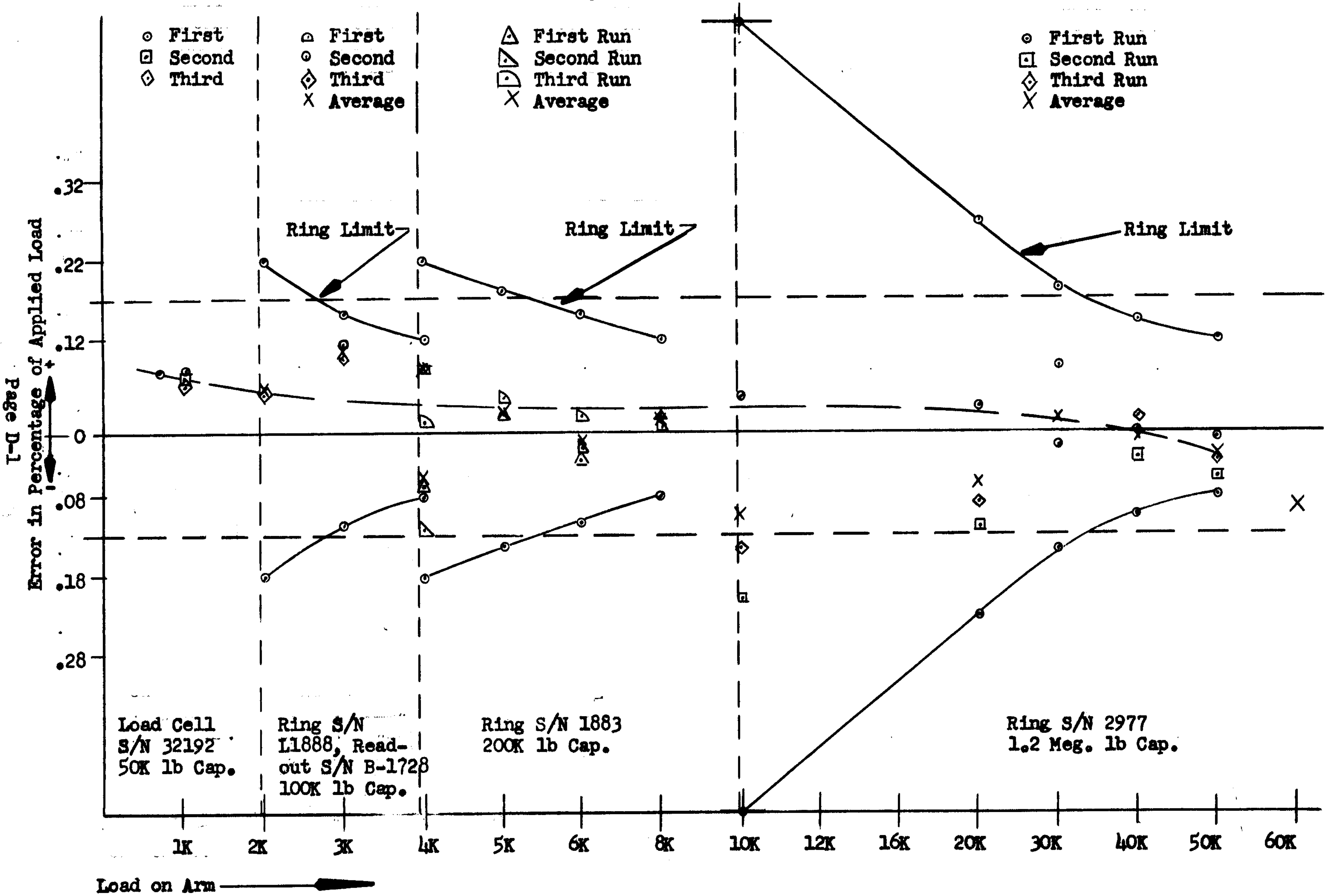
24.975:1
24.9625:1
24.950:1
1K
2K
3K
4K
5K
6K
8K
10K
12K
16K
20K
30K
40K
50K
60K

APPENDIX D

CALIBRATION RESULTS OF PROVING RING METHOD

FORCE MULTIPLYING SYSTEM VS. PROVING RINGS



C O P Y

DAR:GTS:pok

UNITED STATES DEPARTMENT OF COMMERCE
WASHINGTON

Lab. No. 6.4/153939
Project No. 3621

NATIONAL BUREAU OF STANDARDS

Trans. Letter 7-17-58
Order No. 15211-G

REPORT

Calibration
of
MOREHOUSE WEIGHING AND CALIBRATING SYSTEM
submitted by
Morehouse Machine Company
1742 Sixth Avenue
York, Pennsylvania

One Morehouse weighing and calibrating system consisting of Morehouse load rings, No. L 1885, capacity 5,000-lb compression and tension; No. L1886, capacity 20,000-lb compression and tension; No. L 1887, capacity 50,000-lb compression and tension; No. L 1888, capacity 100,000-lb compression and tension; No. L 1889, capacity 300,000-lb compression; and Morehouse balancing instruments Nos. B-1728, B-1861 and B-1862 was calibrated at the National Bureau of Standards. The calibration was completed on July 31, 1958.

The instrument readings were obtained from the micrometer dial when it had been adjusted so that the instrument meter indicated zero. Before the calibration was begun the "zero set" control was set to its midposition and was not adjusted during the calibration. Readings were taken with the sensitivity switch in the "Hi" position.

The rings were loaded in the 10,100-lb and 111,000-lb capacity deadweight testing machines for loads not exceeding 110,000-lb. Loads exceeding 110,000-lb were applied by a hydraulic testing machine and were measured by means of a combination of three Morehouse proving rings, each of 100,000-lb capacity in compression. These rings were calibrated by deadweights in June 1958, and complied with the National Bureau of Standards Specifications for Proving Rings, Letter Circular 822, a copy of which is enclosed. The errors of the applied loads did not exceed 0.02 percent for loads not exceeding 110,000-lb and 0.1 percent for loads greater than 110,000-lb.

The results of the calibration are given in Tables 1 to 15. Values appearing in the tables are quoted to a somewhat greater number of figures than is justified by the overall precision of the measurements in order to show the relative repeatability of the system.

Table 4 - Calibration of Morehouse Load Rings No. L 1888 with Morehouse Balancing Instrument No. B-1728

Applied Load	Change in micrometer reading					
	Tension			Compression		
	Run 1	Run 2	Run 3	Run 1	Run 2	Run 3
lb	div	div	div	div	div	div
10,000	70.73	70.70	70.75	70.73	70.82	70.80
20,000	141.23	141.22	141.28	141.82	141.88	141.93
30,000	211.75	211.75	211.70	212.85	212.97	213.03
40,000	282.10	282.10	282.10	283.87	284.05	284.00
50,000	352.20	352.18	352.20	355.15	355.30	355.28
60,000	422.40	422.30	422.35	426.65	426.70	426.65
70,000	492.38	492.52	492.48	498.22	498.25	498.30
80,000	562.18	562.30	562.25	569.92	570.05	570.10
90,000	632.10	632.08	632.13	641.95	642.18	642.00
100,000	701.68	701.67	701.77	714.10	714.13	714.03

Temperature during calibration = 70°F

C O P Y

UNITED STATES DEPARTMENT OF COMMERCE
WASHINGTON

NATIONAL BUREAU OF STANDARDS

REPORT

JHT:JMS:mln

Lab. No. 6.4/154026
Project No. 3621
Trans. letter 7-25-58
Order No. 15249-G

Calibration
of
MOREHOUSE PROVING RING NO. 1883
submitted by
Morehouse Machine Company
1742 Sixth Avenue
York, Pennsylvania

Proving ring No. 1883, capacity 200,000 lb compression and tension, was submitted by the Morehouse Machine Company for calibration.

The ring was calibrated for compliance with the National Bureau of Standards Specification for Proving Rings, Letter Circular 822, as outlined in Section II.

The ring was calibrated by dead weights up to 111,000 lb. For loads exceeding 111,000 lb in compression, the ring was calibrated by means of a combination of three 100,000-lb capacity proving rings, each of which had been calibrated by dead weights. The results of the calibration are given in the table.

The deflections are correct for a temperature of 70°F. For a temperature of t° F, values of the deflection should be corrected for temperature using the formula

$$d_{70} = d_t - 0.00015\ (t - 70)\ d_t$$

where d_{70} = deflection at a temperature of 70° F

d_t = deflection at a temperature of t° F

t = temperature, degrees Fahrenheit, during test.

The constant, - 0.00015, in the above formula was supplied by the Morehouse Machine Company.

Lab. No. 6.4/154026

Table 1 - Calibration of Morehouse Proving Ring No. 1883, in Compression.

	Deflection of ring					
Applied load	Average of three runs	Run 1 minus avg	Run 2 minus avg	Run 3 minus avg	Avg defl before disassembly minus avg defl after disassembly	Tolerance given by LC 822
lb	div	div	div	div	div	div
20,000	73.05	-0.05	0.0	+0.05	-0.09	±0.37
40,000	146.12	0.0	0.0	0.0	-0.03	±0.74
60,000	219.48	+0.07	-0.01	-0.05	+0.11	±0.74
80,000	292.89	+0.01	+0.03	-0.04	-0.05	±0.74
100,000	366.34	+0.01	+0.08	-0.09	-0.05	±0.74
110,000	403.19	-0.11	+0.06	+0.06	+0.07	"0.74
140,000	513.70	+0.07	-0.05	-0.03	-0.25	±0.74
160,000	587.34	+0.08	-0.14	+0.07	-0.09	±0.74
180,000	661.56	-0.04	-0.11	+0.16	-0.01	±0.74
200,000	735.53	-0.08	-0.07	+0.15	+0.03	±0.74

C O P Y

U.S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS
WASHINGTON 25, D.C.

JHT:JDP:nm

Lab No. 6.4/177461
Project No. 06629
Trans. letter 7-30-63
Order No. 21590-G

NATIONAL BUREAU OF STANDARDS
REPORT OF CALIBRATION

MOREHOUSE PROVING RING NO. 2977
Submitted by
Morehouse Machine Company
York, Pennsylvania

Proving ring No. 2977, capacity 1,200,000 lb compression and tension, was submitted by the Morehouse Machine Company with a request that it be calibrated in compression only.

The ring was calibrated in compression in accordance with Section II of the Appendix to Circular 454 of the National Bureau of Standards.

Loads not exceeding 300,000 lb were applied by a mechanical testing machine and measured by means of a combination of three 100,000-lb capacity proving rings, each of which had been calibrated by dead weights. Loads exceeding 300,000 lb but not exceeding 900,000 lb were applied by a hydraulic testing machine and measured by means of a combination of three 300,000-lb capacity proving rings, each of which had been calibrated by dead weights and a combination of three 100,000-lb capacity proving rings. Loads exceeding 900,000 lb were applied by a hydraulic testing machine and measured by means of a combination of five 300,000-lb capacity proving rings, each of which had been calibrated by dead weights and a combination of three 100,000-lb capacity proving rings. The errors of the applied loads did not exceed 0.1 percent. The results of the calibration are given in Table 1.

The deflections are correct for a temperature of 70° F. For a temperature of t° F, the values of the deflection should be corrected for temperature using the formula

$$d_{70} = d_t - 0.00015\,(t - 70)d_t$$

where d_{70} = deflection at a temperature of 70° F

d_t = deflection at a temperature of t° F

t = temperature, degrees Fahrenheit, during test.

The constant, -0.00015, in the above formula was supplied by the Morehouse Machine Company.

C O P Y

Lab No. 6.4/177461

Table 1 - Calibration of Morehouse Proving Ring No. 2977 in Compression

Load	Deflection of ring after disassembly				Avg defl before disassembly minus avg after disassembly	Tolerance given by Appendix to C454
	Average of three runs	Run 1 minus avg	Run 2 minus avg	Run 3 minus avg		
lb	div	div	div	div	div	div
120,000	75.31	-0.05	+0.05	0.00	-0.02	±0.38
240,000	150.64	+0.05	-0.07	+0.02	-0.01	±0.75
360,000	226.18	-0.03	+0.02	+0.02	+0.01	±0.75
480,000	301.58	-0.04	+0.01	+0.03	-0.07	±0.75
600,000	377.01	+0.02	+0.02	-0.03	-0.28	±0.75
720,000	452.46	-0.10	-0.05	+0.15	-0.15	±0.75
840,000	527.90	+0.09	-0.16	+0.06	-0.07	±0.75
960,000	603.39	+0.10	+0.01	-0.10	-0.06	±0.75
1,080,000	679.08	+0.07	+0.08	-0.14	-0.06	±0.75
1,200,000	754.72	-0.15	-0.02	+0.16	-0.17	±0.75

This ring complies with the requirements of Section II of the Appendix to Circular 454 for compression loads. Because the ring was not calibrated in tension, the results are given in tabular form. An excerpt from the Appendix is enclosed.

For the Director,

s/L. K. Irwin

L. K. Irwin, Chief
Engineering Mechanics Section
Division of Mechanics

Enclosure

Washington, D. C.

APPENDIX E

BALDWIN-LIMA-HAMILTON CALIBRATION
METHOD FOR LOAD CELLS

C O P Y

BALDWIN-LIMA-HAMILTON CORPORATION

ELECTRONICS DIVISION

WALTHAM 54, MASS.

CALIBRATION CERTIFICATES

FOR

TWO (2) 1,500,000 POUND CAPACITY LOAD CELLS

TYPE U3XXA

LOAD CELL CATALOG NO. 271946

LOAD CELL SERIAL NUMBERS: 32459
32460

BLH JOB NO. 050-3595

AGC P. O. NO. A-170028

C O P Y

SPECIFICATIONS

The load cells shipped on this order were to the following AGC specifications as referenced on Purchase Order A-170028, 3/26/63.

Special U3XXA, double bridge, Catalog No. 271946, 0-1.5 million pound cells standardized in tension per AGC Specification No. 32003/1 with the following exceptions:

Item 3.3.4.12 - Linearity: 0.15% F.S. in tension

Item 3.3.4.13 - Hysteresis: 0.15% F.S. in tension

Item 3.3.4.15 - Combined Effect: 0.30% F.S.

Item 3.3.4.16.1 - For 1° offset, error shall not exceed 0.2% F.S.

Item 3.3.4.16.2 - For 3° offset, error shall not exceed 0.5% F.S.

Item 3.3.4.17 - Insulation Resistance: 1000 megohms at 50 VDC at any temperature 30°F to 130°F.

Item 3.3.4.18 - Maximum Deflection: 0.040"

METHOD OF CALIBRATION

The load cells were calibrated on a 2,400,000 lb capacity Tension-Compression Testing Machine at the Watertown Arsenal, Watertown, Mass. The machine was calibrated in February, 1963 by the Wiedemann Machine Company, using five (5) 200,000 pound and one (1) 1,200,000 pound proving rings rented from the Morehouse Machine Company. The N.B.S. Calibration Data on the 1,200,000 pound proving ring is enclosed. Also, enclosed is a descriptive sheet on the testing machine.

CALIBRATION PROCEDURE

It was found that the most accurate operating region of the testing machine was the 0 to 1,200,000 pound range. This was determined by analyzing the test cell linearity and repeatability data and cross-checking six (6) load points within the range, using an 800,000 pound BLH shop standard load cell. The calibration data, as presented on Form 609, was generated as follows:

Tension, Linearity and Hysteresis

All loads up to 80% (1,200,000 pounds) of full scale were read directly on the 1,200,000 pound testing machine dial range. The 96% and 100% full scale points were read directly on the 2,400,000 pound testing machine dial range. The data, as presented, is the average of three (3) successive runs. No load corrections or extrapolations were made to arrive at the output data.

Compression F.S. Output

The full scale compression output was determined by a linear extrapolation of the actual output at an uncorrected load of 1,200,000 pounds. Three (3) runs were made to verify the repeatability of the 80% load point.

Shunt To Force

The shunt to force loadings were all done in tension from 30°F to 130°F. The Absolute Full Scale Output, MV/V, was determined by a linear extrapolation of the actual output at an uncorrected load of 1,200,000 pounds. Proportional parts of the calculated full scale (1,500,000 pounds) output were used to verify the shunt to force requirement.

Sensitivity

The sensitivity (full scale output) was adjusted in tension at a load of 1,200,000 pounds. The output was set at 2.4 millivolts per volt at this load point.

C O P Y

MOREHOUSE
MACHINE COMPANY

INSTRUMENT MAKERS
TOOL MAKERS

1742 SIXTH AVENUE
YORK, PENNSYLVANIA, 17403
Telephone 30081

October 17, 1963

Baldwin-Lima-Hamilton Corporation
Electronics Division
Waltham 54, Massachusetts

Attn: Mr. B. H. Shapiro, Product Engineer, Transducers

Gentlemen:

In accordance with your letter of October 15, we are pleased to enclose information about our proving rings. We are also enclosing N.B.S. calibration data for the 1,200,000 lbs. capacity proving ring that was used to calibrate the Watertown Arsenal testing machine.

May we hear from you if any further information is desired?

Sincerely yours,

MOREHOUSE MACHINE COMPANY

s/Henry A. Zumbrun

Henry A. Zumbrun

HAZ/jj

Encl: 159, 204,
photo-copy for S/N 2545

THE MOREHOUSE PROVING RING
AND OTHER INSTRUMENTS FOR CALIBRATING PHYSICAL TESTING MACHINES AND LOAD MEASURING SYSTEMS

C O P Y

UNITED STATES DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS
WASHINGTON 25, C.D.

Lab. No. 6.4/167203
Project No. 06629
Trans. letter 6-9-61
Order No. 19018-G

NATIONAL BUREAU OF STANDARDS

REPORT OF CALIBRATION

MOREHOUSE PROVING RING NO. 2545
submitted by

Morehouse Machine Company
1742 Sixth Avenue
York, Pennsylvania

Proving ring No. 2545, capacity 1,200,000 lb compression and tension, was submitted by the Morehouse Machine Company for calibration.

The ring was calibrated for compliance with the National Bureau of Standards Specification for Proving Rings, Letter Circular 822, as outlined in Section II.

Loads not exceeding 300,000 lb were applied by a mechanical testing machine and measured by means of a combination of three 100,000-lb capacity proving rings, each of which had been calibrated by dead-weights. Loads exceeding 300,000 lb but not exceeding 600,000 lb were applied by a mechanical testing machine and measured by means of a combination of three 300,000-lb capacity proving rings, each of which had been calibrated by dead-weights and a combination of three 100,000-lb capacity proving rings. Loads exceeding 600,000 lb were applied by a hydraulic testing machine and measured by means of a combination of four 300,000-lb capacity proving rings, each of which had been calibrated by dead-weights and a combination of three 100,000-lb capacity proving rings. The errors of the applied loads did not exceed 0.1 percent. The proving rings used to measure the loads applied during this calibration were calibrated in May and June 1961 and complied with the National Bureau of Standards Specification for Proving Rings, Letter Circular 822, a copy of which is enclosed. The results of the calibration are given in the table.

The deflections are correct for a temperature of 70° F. For a temperature of t° F, the values of the deflection should be corrected for temperature using the formula

$$d_{70} = d_t - 0.00015\,(t - 70)\,d_t$$

where d_{70} = deflection at a temperature of 70°

d_t = deflection at a temperature of t° F

t = temperature, degrees Fahrenheit, during test

The constant, -0.00015, in the above formula was supplied by the Morehouse Machine Company.

C O P Y

Lab No. 6.4/167203

Table - Calibration of Morehouse Proving Ring No. 2545

Load	Deflection of ring after disassembly: Average of three runs	Run 1 minus avg	Run 2 minus avg	Run 3 minus avg	Avg defl before disassembly minus avg after disassembly	Tolerance given by LC 822
lb	div	div	div	div	div	div
120,000	75.25	-0.03	-0.02	+0.05	-0.03	±0.38
240,000	150.38	+0.07	-0.09	+0.03	+0.07	±0.75
300,000	187.97	-0.03	-0.05	+0.07	+0.01	±0.75
450,000	282.10	+0.15	-0.04	-0.12	-0.03	±0.75
600,000	376.25	+0.09	-0.07	-0.03	+0.03	±0.75
720,000	451.78	-0.09	+0.11	-0.02	-0.02	±0.75
840,000	526.90	+0.03	-0.01	-0.01	+0.13	±0.75
960,000	602.54	+0.05	-0.07	+0.03	+0.07	±0.75
1,080,000	678.28	-0.02	+0.03	0.0	-0.11	±0.75
1,200,000	753.52	+0.13	-0.05	-0.08	+0.17	±0.75

This ring complies with all the requirements of LC 822 for compression loads. Because the ring was not calibrated in tension, the results are given in the form of a report instead of a certificate.

For the Director,

s/L. K. Irwin

L. K. Irwin, Chief
Engineering Mechanics Section
Division of Mechanics

Enclosure

Washington, D. C.

COPY

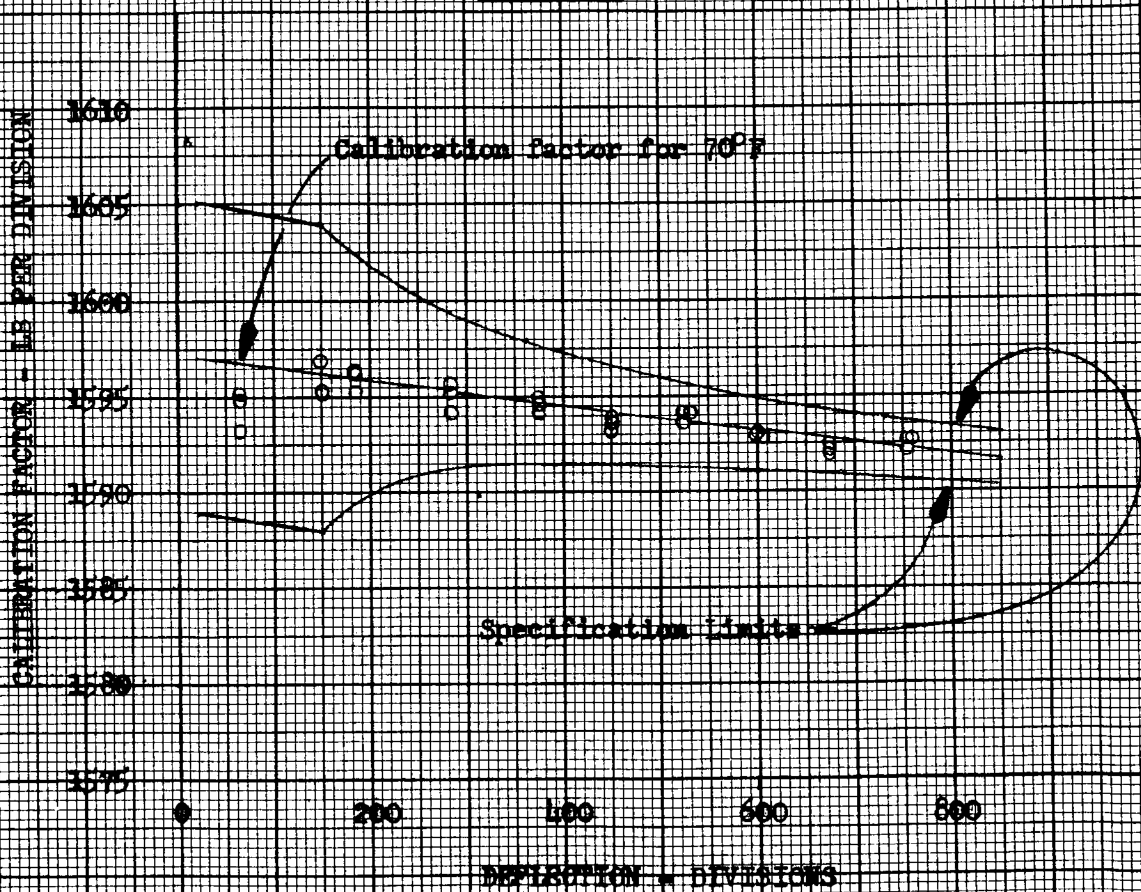
CALIBRATION GRAPH

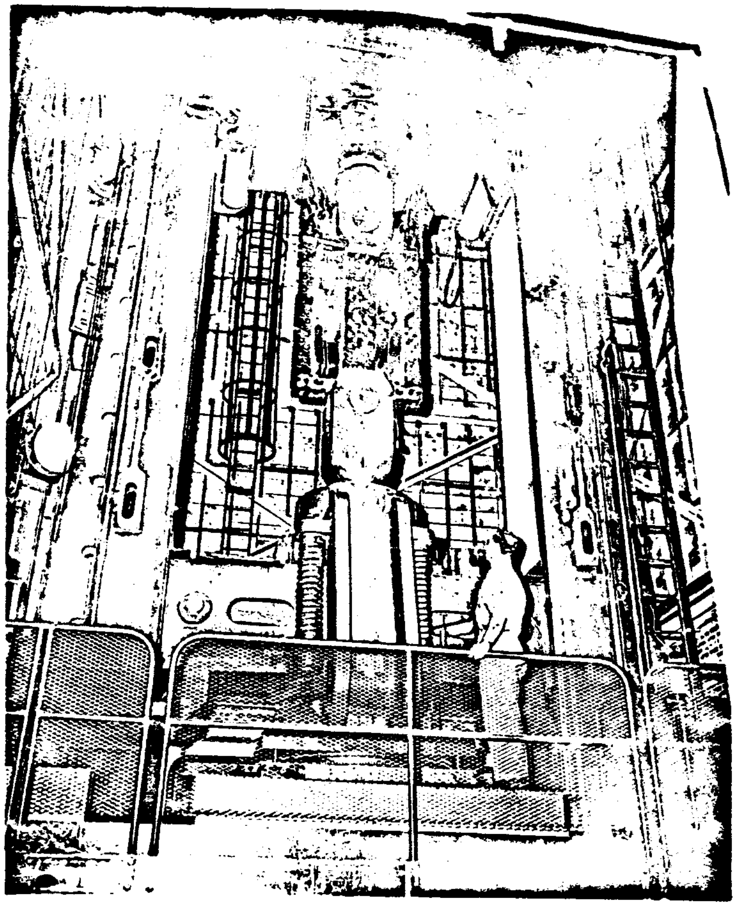
Proving Ring No. 2545

Capacity 1,200,000 lb July 24, [illegible]

Compression

MOREHOUSE MACHINE CO.
York, Pa.





WTN.681 - 1022 2,400,000 lbs. CAPACITY
TENSION-COMPRESSION TESTING MACHINE

TYPE: Baldwin-Lima-Hamilton, Hydraulic

TESTING SPEEDS: 0.002"/min. to 4.00"/min. Return Speed: 15"/min.

MATERIALS OVERHEAD CRANE CAPACITY: One Hook = 2 ton, One Hook = 5 ton.

OPENING BETWEEN PLATENS: Tension = 15 ft. maximum
Compression = 20 ft. maximum

PISTON TRAVEL: 5 ft.

SPECIMENS AND HOLDERS: TENSION:

1. Vee Grips
   a. Maximum = 6" diameter x 20' length
   b. Minimum = 4" diameter x 4' length

2. Threaded Grips
   Available Holder = 5.00" diameter - 4 threads/inch

3. Pin Hole Grips
   a. Maximum = 10.5" th x 10.0' wide x 12.0' long using a 9.0" diameter pin
   b. Minimum = Any desired size but side shims and pin bushings will have to be fabricated

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